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MATHEMATICAL THEORY OF

THE STEAM ENGINE

THE MATHEMATICAL THEORY OF

THE STEAM ENGINE

WITH RULES AT LENGTH, AND EXAMPLES WORKED OUT FOR THE USE OF PRACTICAL MEN

By T. BAKER, C.E.

AUTHOR OF "STATICS AND DYNAMICS," "LAND AND ENGINEERING SURVEYING," "ELEMENTS OF MECHANISM," "MINING SURVEYING," ETC.

With numerous Diagrams

SIXTH EDITION, REVISED BY PROFESSOR J. R. YOUNG

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LUDGATE HILL

1875

PREFACE.

A WORK of this kind, on the Mathematical Theory of the Stationary, Marine, and Locomotive Engines, has long been a desideratum; not only as an introduction to Tredgold's large and important work on the same subject, but also for the use of a numerous class of students, who either have not time to read or the means of purchasing the large work just referred to. The author of this "Introduction" has taken great pains to supply this link in the chain of scientific research so much required, as well as to adapt it to the wants of practical men, by giving rules in words at length for their use; also for students who have not yet accustomed themselves to the application of mathematical formulæ, by which their progress in studies of this kind will be greatly facilitated, until at length they arrive at full competence in both the theoretical and practical parts of these important subjects, and thus be prepared to understand with ease the various complexities of Tredgold's large and complete work.

The author has, moreover, given in this "Introduction" a new and important method of showing the positions of the slide-valves, corresponding to the various openings of the steam and exhausting ports, by geometrical construction,

in addition to the rules usually given by calculation, which method has been presented to him by the eminent engineer, C. E. Amos, Esq., M.I.C.E., of the well-known firm of Easton, Amos, and Sons. It is given in as clear a manner as the subject seems to admit of, since any intelligent workman may, with rule and compasses, draw the figure to any convenient scale, and as respects locomotive engines, he may draw it to the full size.

The work concludes with an Appendix, on the strength, &c., of several important parts of the Steam Engine; to which are added Tables of Hyperbolic Logarithms, and Tables of Friction, which are used in the calculations in the preceding parts of the work.

T. BAKER

CONTENTS.

THE

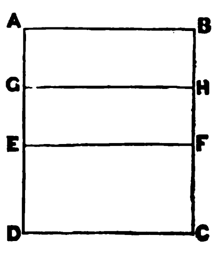
MATHEMATICAL THEORY

OF THE

STEAM ENGINE.

THE preliminary step in this important subject is to treat of the law of the expansion of steam, which was discovered by Mariotte, and confirmed by Arago and Dulong, to all pressures as far as up to 27 atmospheres. *This law is the simplest that can be applied to the steam engine;* it may be enunciated thus:—If a given weight of steam be made to vary its volume without changing its temperature, the *elastic force* of the steam will vary in the *inverse ratio of the volume* it is made to occupy.

Thus let A B C D be a cylinder, E F and G H any two positions of the piston; then the pressure in the position E F is to the pressure in the position G H as the space G H C D is to the space E F C D.



Although this is by far the most simple and general law that we can apply, yet it is not strictly correct; but sufficiently so for the purpose of showing the great advantages that can be gained by working steam expansively; the results, however, will be alway a little greater than the truth, since *during the time the expansion is going on in the cylinder, the temperature does not remain exactly the same;* therefore, the various properties of steam given by Arago, Dulong, Tredgold, Pambour, Pole, and others, shall hereafter be produced in detail.

B

UNIT OF WORK.

In applying this principle to estimate and compare the different kinds of work performed under different circumstances, it becomes necessary to have a distinct measure or *unit of work*, by which the various results may be calculated and the amounts ascertained.

The English unit of work is the power necessary to raise one pound through a space of one foot.

Thus, if 1 lb. be raised 1 foot by a machine, then one unit of work has been performed; if 4 lbs. be raised 6 feet, then $4 \times 6 = 24$ units of work have been performed; and so on for other combinations. *Hence the units of work performed are measured by the product of the weight of the body in pounds, and the space or height in feet, through which it is raised;* also pressures or resistances of every kind, in whatever direction they are exerted, may be expressed in pounds, and therefore measured by the unit of work here described.

Let this unit of work be referred to one minute, as the unit of time. Taken in this point of view the unit of work will be represented as 1 lb. raised 1 foot high in 1 minute. It is assumed that a horse is able to raise 33,000 lbs. one foot high in one minute; and this is, therefore, called a horse-power. Hence, to find the number of horse-powers consumed in any proposed work, it will only be required to divide the units in that work by 33,000 times the number of minutes in which it is done.

To find the units of work done on the piston in one stroke.

Given the length of the stroke of the piston (l), the distance moved by the piston before the steam is cut off (q), and the pressure at which the steam is admitted to the cylinder (p); to find the work done upon each square inch of the piston in one stroke.

The required work on each square inch of the piston is, by Mariotte's law,

$$q\,p\left(1 + \log \frac{l}{q}\right)^*$$

* The logarithms here used are the hyperbolic; the investigation of this and the other formulæ will be hereafter given.

But that this Treatise may be useful to such practical men as are not accustomed to use formulæ, the following rule is given :—

RULE 1.—Divide the whole length of the stroke by the distance moved by the piston before the steam is cut off, and find the hyperbolic logarithm of the quotient, to which add 1, and multiply the sum by the product of the pressure of the steam and the distance moved by the piston before the steam is cut off, and the result is the whole work done on each square inch of the piston.

To find the load.

The load is generally defined to be the mean pressure of the steam.

Let L be the load; then

$$L = \frac{qp}{l}\left(1 + \log.\frac{l}{q}\right).$$

Or, in words at length, we have the following :—

RULE 2.—Divide the work done upon one square inch in one stroke by the length of the stroke, and the quotient will be the load.

To find the pressure at which the steam is admitted.

$$p = \frac{L.l}{q\left(1 + \log\frac{l}{q}\right)}.$$

Or, in words at length, there results the following :—

RULE 3.—Divide the whole length of the stroke by the number of feet described by the piston before the steam is cut off; to the hyperbolic logarithm of this quotient add unity, and multiply the sum by the number of feet moved by the piston before the steam is cut off, for a divisor; and for a dividend multiply the length of the stroke by the load. The quotient is the pressure in pounds per square inch at which the steam is admitted.

To find the horse-powers.

Let A = area of the piston in square inches, and N = number of strokes per minute; then,

$$\text{Horse-power} = \frac{\text{A N} \, q \, p \left(1 + \log \frac{l}{q}\right)}{33{,}000}.$$

Or, in words at length, we have the following :—

RULE 4.—Multiply the work done on one square inch in one stroke, by the area of the piston in square inches, and by the number of strokes per minute, and this product being divided by 33,000, gives the horse-powers.

Example 1.—The pressure of the steam upon the piston is 60 lbs. per square inch, the length of the stroke is 12 feet, the steam is cut off at $\frac{1}{6}$ of the stroke; required the number of units of work upon each square inch of the piston, and the horse-powers, the diameter of the cylinder being 50 inches, and the piston making 20 strokes per minute.

By Rule 1,

$$\log \frac{12}{2} = \log 6 = 1{\cdot}79175, \text{ and}$$

$$1{\cdot}79175 + 1 = 2{\cdot}79175,$$

$\therefore 2 \times 60 \times 2{\cdot}79175 = 335{\cdot}01 =$ whole work done on one square inch of the piston in one stroke.

Again, $50 \times 50 \times {\cdot}7854 = 1963{\cdot}5$ area of the piston in square inches, and by Rule 4,

$$\frac{335{\cdot}01 \times 1963{\cdot}5 \times 20}{33{,}000} = 398{\cdot}66 \text{ horse-powers.}$$

By the formula $q \, p \left(1 + \log \frac{l}{q}\right) = 2 \times 60 \, (1 + 1{\cdot}79175)$

$= 335{\cdot}01 =$ whole work done on one square inch of the piston in one stroke.

Also,

$$\frac{\text{A N} \, q \, p \left(1 + \log \frac{l}{q}\right)}{33{,}000} = \frac{1963{\cdot}5 \times 20 \times 2 \times 60 \, (1 + 1{\cdot}79175)}{33{,}000} =$$

$$398{\cdot}66 \text{ horse-powers.}$$

Example 2.—The things given being the same as in Example 1, it is required to find the load of the engine.

Here the work done on one square inch in one stroke of

the piston, is 335·01 units; whence $\frac{335 \cdot 01}{12} = 27 \cdot 9175 =$ load in lbs. on one square inch of the piston, by Rule 2.

Example 3.—Required the pressure per square inch at which the steam must be admitted, when the load is 44 lbs. per square inch; the steam is cut off at 2 feet, and the length of the stroke 5 feet.

By Rule 3,

$$1 + \log 2 \cdot 5 = 1 + \cdot 9163 = 1 \cdot 9163,$$

$$2 \times 1 \cdot 9163 = 3 \cdot 8326, \text{ and}$$

$$44 \times 5 \div 3 \cdot 8326 = 57 \cdot 4 \text{ lbs. per square inch nearly.}$$

Example 4.—The pressure of steam upon the piston is 60 lbs. per square inch; the resistance from imperfect condensation 4 lbs. per square inch; the length of the stroke is 12 feet, and the steam is cut off at $\frac{1}{6}$ of the stroke. Required, the number of units of work done upon each square inch of the piston, and the number of units of work gained by working the steam expansively,

Here, as in Example 1, 335·01 units of work are done on one square inch of the piston in one stroke.

But, as the resistance from uncondensed vapour is 4 lbs. per square inch, it will be $12 \times 4 = 48$ lbs. for the resistance for the whole length of the stroke; therefore the effective work will be $335 \cdot 01 - 48 \cdot 00 = 287 \cdot 01$ units of work on one square inch.

Now, to find the advantage gained by working the steam expansively; we have when the engine works without expansion $12 \times 60 = 720 =$ units of work done upon each square inch.

But, as in this case the steam is not cut off till the end of the stroke, there is $\frac{12}{2} = 6$ times the steam used; hence for the same quantity there results,

$$\frac{720}{6} = 120 \text{ lbs. per square inch.}$$

While by working expansively there has been obtained 335·01 units of work; hence $335 \cdot 01 - 120 = 215 \cdot 01$ lbs. gained in this example.

Also, $\frac{335 \cdot 01}{120} = 2 \cdot 79$ times as much work done by the same quantity of steam when worked expansively.

When it is stated that the steam is cut off at $\frac{1}{6}$, $\frac{1}{5}$, $\frac{1}{4}$, &c., of the stroke, there is no necessity for dividing the whole distance moved by the piston by the distance it moves before expansion, whatever may be the fractional part; we need only find the hyperbolic logarithm of the denominator of the fraction, and add unit to it; then multiply the sum by the product of the pressure and the part of the stroke before expansion begins, and the last product is the whole work done upon each square inch of the piston. For example, let the length of the stroke be 12 feet, and the steam be cut off at $\frac{1}{3}$ of the stroke, the pressure upon the piston being 60 lbs. per square inch.

Here the denominator of the fraction is 3, and its hyperbolic logarithm is 1·0986, to which add unity, and we shall have 2·0986; whence $4 \times 60 \times 2 \cdot 0986 = 503 \cdot 664 =$ whole work done upon each square inch of the piston in one stroke.

To find the work done by expansion without the use of logarithms.

Rule.—Divide that part of the stroke through which expansion takes place into any even number of equal parts, and calculate the pressure per square inch upon the piston at each division of the stroke by Mariotte's law; take the sum of the extreme pressures in pounds per square inch, four times the sum of the even pressures, and twice the sum of the odd pressures; multiply the sum of all these by one-third of the common distance between the positions of the piston, and the result will be the work done upon each square inch of the piston after expansion begins. The work done before expansion begins being evidently equal to the pressure per square inch multiplied by the number of feet moved before expansion; and the whole work done during a single stroke is equal to the sum of the works done before and after expansion.

Example 1.—The pressure of the steam upon the piston is 80 lbs. per square inch, the length of the stroke of the *piston is* 10 feet, the steam is cut off at $\frac{1}{5}$ of the stroke; it is

required to find the whole amount of work done upon each square inch of the piston.

The piston moves $\frac{1}{5}$ of the stroke, which is 2 feet, with the full pressure of the steam; let the remaining part of the stroke, which is 8 feet, be divided into four equal parts, each of which will be 2 feet; let these pressures be represented by P, P_1, P_2, &c.; then, by Mariotte's law,

$$4 : 2 :: 80 : P_1 = \frac{80 \times 2}{4} = 40 \text{ lbs. pressure.}$$

$$6 : 2 :: 80 : P_2 = \frac{80 \times 2}{6} = 26 \cdot 66 \text{ lbs.} \quad ,,$$

$$8 : 2 :: 80 : P_3 = \frac{80 \times 2}{8} = 20 \text{ lbs.} \quad ,,$$

$$10 : 2 :: 80 : P_4 = \frac{80 \times 2}{10} = 16 \text{ lbs.} \quad ,,$$

Then, by the rule just given,

80 = greatest extreme pressure.
16 = least extreme pressure.

96 = sum of extreme pressures.

40
20

60 = sum of the even pressures.
4

240 = 4 times the sum of the even pressures

26·66′ = the odd pressure, there being only one
2

53·33′ = twice the odd pressure.

96 = sum of extreme pressures.
240 = 4 times sum of even pressures.
53·33′ = twice the odd pressure.

389·33′ = sum.

The above sum, *i.e.*, 389·33 being multiplied by ⅓ of 2 feet, *the common* distance, gives

$$\frac{389 \cdot 33 \times 2}{3} = 259 \cdot 55 =$$

the number of units of work done upon each square inch of the piston after expansion begins, and $80 \times 2 = 160 =$ the number of units of work done upon each square inch before expansion begins. Hence

$$\begin{array}{r} 259 \cdot 55 \\ 160 \\ \hline 419 \cdot 55 = \end{array}$$

number of units of work done upon each square inch, during a single stroke of the piston.

Example 2.—The pressure of steam upon the piston is 60 lbs. per square inch, the resistance arising from imperfect condensation, 4 lbs. per square inch, the length of the stroke is 12 feet, and the steam is cut off at $\frac{1}{6}$ of the stroke; it is required to determine the number of units of work done upon each square inch of the piston, the number of units of work gained by working expansively, the load per square inch, and the position of the piston when the velocity is greatest.

Let the remaining part of the stroke, *i. e.* $12 - 2 = 10$ feet be divided into 10 equal parts; then

$$3 : 2 :: 60 : P_1 = \frac{60 \times 2}{3} = 40,$$

$$\text{and similarly } P_2 = \frac{60 \times 2}{4} = 30,$$

$$P_3 = \frac{60 \times 2}{5} = 24,$$

$$P_4 = \frac{60 \times 2}{6} = 20,$$

$$P_5 = \frac{60 \times 2}{7} = 17 \cdot 142,$$

$$P_6 = \frac{60 \times 2}{8} = 15,$$

$$P_7 = \frac{60 \times 2}{9} = 13{\cdot}333,$$

$$P_8 = \frac{60 \times 2}{10} = 12,$$

$$P_9 = \frac{60 \times 2}{11} = 10{\cdot}909,$$

$$P_{10} = \frac{60 \times 2}{12} = 10,$$

60 + 10 = 70 = sum of extreme pressures.

```
 40·000
 24·000
 17·142
 13·333
 10·909
───────
105·384 = the sum of the even pressures.
      4
───────
421·536 = 4 times the sum of the even pressures.

 30·000
 20·000
 15·000
 12·000
───────
 77·000 = the sum of the odd pressures.
      2
───────
154·000 = twice the sum of the odd pressures.
421·536 = 4 times the sum of the even pressures.
 70·000 = sum of the extreme pressures.
───────
3)645·536
───────
215·178
120     = 60 × 2 = work done before expansion.
───────
335·178 = whole work done upon each square inch.
```

By the formula, the whole work done per stroke is

$$= qp\left(1+\log\frac{l}{q}\right) = 2 \times 60\left(1 + \log\frac{12}{2}\right)$$
$$= 120\,(1 + \log 6) = 120\,(1 + 1{\cdot}79175)$$
$$= 120 \times 2{\cdot}79175 = 335{\cdot}01,$$

which is very nearly the same as that deduced by the rule.

Since the resistance from uncondensed vapour is 4 lbs. per square inch, then $12 \times 4 = 48$ lbs. = whole resistance, and by subtracting this from 335·178, the whole work done per square inch, there will remain $335{\cdot}178 - 48 = 287{\cdot}178$ for the effective work.

To find the advantage derived from working steam expansively.

When the steam works without expansion, then $12 \times 60 = 720$ = work done upon each square inch; but as the steam is cut off at $\frac{1}{6}$ of the stroke in working it expansively, there is only $\frac{1}{6}$ of the quantity of steam used in this case, or $\frac{720}{6} = 120$ lbs. per square inch, and $335{\cdot}178 - 120 = 215{\cdot}178$ lbs. gained in this case; or $\frac{335{\cdot}178}{120}$ = nearly 2·8 times as much work done by the same quantity of steam, when worked expansively.

The work required to move the load to the end of the stroke is 12 L, and the work of the steam has been expended to produce this effect;

$$\therefore 12\,\mathrm{L} = 335{\cdot}178$$

$$\text{or, } \mathrm{L} = \frac{335{\cdot}178}{12} = 27{\cdot}9315 \text{ lbs. per square inch.}$$

If s be the distance moved over by the piston, when it attains its greatest velocity, then by Mariotte's law,

$$s : 2 :: 60 : 27{\cdot}9315,$$

$$\textit{whence } s = \frac{60 \times 2}{27{\cdot}9315} = 4{\cdot}3.$$

By the formula,

$$s = \frac{l}{1 + \log \frac{l}{q}} = \frac{12}{1 + \log 6} = \frac{12}{2{\cdot}79176} = 4{\cdot}3$$

nearly, the same as above.

Example 3.—To find the pressure per square inch at which the steam must be admitted when the load is 22 lbs. per square inch, the length of the stroke 5 feet, and the steam is cut off at 2 feet; the resistance from uncondensed vapour being 2 lbs. per square inch.

Let k = pressure per square inch; then $P = x$, and by taking the distances at every $\frac{1}{2}$ foot beyond 2 feet, where the steam is cut off, there will result

$$\frac{5}{2} : 2 :: x : P_1,$$

$$\therefore P_1 = \frac{4x}{5};$$

and by proceeding in a similar manner, the successive pressures for every $\frac{1}{2}$ inch of the remaining 3 inches, there will result,

$$P_2 = \frac{2x}{3},\ P_3 = \frac{4x}{7},\ P_4 = \frac{x}{2},\ P_5 = \frac{4x}{9},\ \text{and } P_6 = \frac{2x}{5},$$

$x + \frac{2}{5}x = \frac{7}{5}x$ = the sum of the greatest and least extreme pressures per square inch;

$4\left(\frac{4x}{5} + \frac{4x}{7} + \frac{4x}{9}\right) = \frac{2288x}{315}$ = 4 times the sum of the even pressures;

and $2\left(\frac{2x}{3} + \frac{x}{2}\right) = \frac{7x}{3}$ = twice the sum of the odd pressures;

$\therefore \frac{2288x}{315} + \frac{7x}{3} + \frac{7x}{5} = \frac{3464x}{315}$ = the sum of all the pressures after expansion. This sum, multiplied by $\frac{1}{3}$ of the intervening space, *i. e.* by $\frac{1}{3}$ of $\frac{1}{2}$ an inch $= \frac{1}{6}$, gives $\frac{1732x}{945}$ = work after expansion, and $2x$ is the work done *before* expansion;

$\therefore 2x + \frac{1732\,x}{945} = 22 \times 5 = 110$, or $x =$ 29 lbs. per square inch nearly.

Note.—In the examples just given, referring to condensing engines, no account has been taken of the friction, clearance of the piston, and other minutiæ; but after the mathematical theory has been given, examples will be added, in which all these resistances will be included, referring both to condensing and non-condensing engines. The object of giving the preceding examples is to enable the student to obtain an idea of the method of speedily arriving at something like an approximation to finding the horse-powers of any proposed condensing engine.

On the Resistances to the Pressure of the Steam in Non-condensing and Condensing Engines.

From the experiments made by Pambour and others, these resistances in non-condensing engines are as follow:—

The pressure of the atmosphere, which produces a resistance of about 15 lbs. per square inch; also the resistance arising from various parts of the engine—at a mean, the estimate is 1 lb. to the square inch for the unloaded engine; and an additional friction of $\frac{1}{7}$ of the effective pressure or useful load, for overcoming the friction of the loaded engine.

Supposing the pressure of the steam to be 90 lbs. per square inch; the resistances are 15 lbs. per square inch from the atmosphere, and 1 lb. for the friction of the unloaded engine; therefore $15 + 1 = 16$ lbs. must be taken from 90, which leaves 74; hence the load $+ \frac{1}{7}$ of the load is equal to 74 lbs., or $\frac{8}{7}$ of the load $= 74$; therefore the load $= \frac{74 \times 7}{8} =$ $64\frac{3}{4}$ lbs. $=$ the effective pressure per square inch on the piston.

Hence the following RULE.—*To find the Load when Resistances are taken into account, viz., The Useful Load.*

From the pressure of the steam in the cylinder subtract 16, multiply the remainder by 7, and divide this product by 8, and the quotient will be the useful load.

Example.—Given the pressure of the steam in the cylinder 60 lbs. per square inch, to find the useful load.

By the rule,

$$\frac{(60 - 16) \times 7}{8} = 38\tfrac{1}{2} \text{ lbs.}$$

For condensing or low-pressure engines, instead of the

resistance of the atmosphere, we must take into account the resistance of the vapour in the condenser, which is generally estimated at 4 lbs. per square inch; in this case there results

$$\text{load} + \tfrac{1}{7}\,\text{load} + 1 + 4 = \text{whole pressure of the steam.}$$

Hence the following RULE.—*To find the Load when Resistances are taken into account, in the Condensing Engine.*

From the mean pressure of the steam subtract 5, and $\frac{7}{8}$ of the remainder is the useful load.

On the Evaporating Power of the Boiler.

The evaporating power of the boiler is of the utmost importance, and as it is the source of all efficient work produced by the steam engine, many very ingenious contrivances have been made to augment this evaporating power; these contrivances will hereafter be explained. The quantity of work done depends on the quantity of water evaporated, as well as on the temperature, and the pressure at which the steam is generated. Formulæ shall hereafter be given to find the relation between the volume of steam and the pressure, but the following experimental table will give results sufficiently correct for all practical purposes; it shows the volume which a cubic foot of water produces in the form of steam at the several different pressures, as well as the corresponding temperatures.

Total pressure in pounds per sq. inch.	Corresponding temperature by Fahrenheit's thermometer.	Volumes of the steam compared to the volume of the water that has produced it.	Total pressure in pounds per sq. inch.	Corresponding temperature by Fahrenheit's thermometer.	Volumes of the steam compared to the volume of the water that has produced it.
1	102·9	20954	14	209·0	1777
2	126·1	10907	15	213·0	1669
3	141·0	7455	16	216·4	1572
4	152·3	5696	17	219·6	1487
5	161·4	4624	18	222·6	1410
6	169·2	3901	19	225·6	1342
7	176·0	3380	20	228·3	1280
8	182·0	2985	21	231·0	1224
9	187·4	2676	22	233·6	1172
10	192·4	2427	23	236·1	1125
11	197·0	2222	24	238·4	1082
12	201·3	2050	25	240·7	1042
13	205·3	1903	26	243·0	1005

Total pressure in pounds per sq. inch.	Corresponding temperature by Fahrenheit's thermometer.	Volumes of the steam compared to the volume of the water that has produced it.	Total pressure in pounds per sq. inch.	Corresponding temperature by Fahrenheit's thermometer.	Volumes of the steam compared to the volume of the water that has produced it.
27	245·1	971	69	303·2	411
28	247·2	939	70	304·2	406
29	249·2	909	71	305·1	401
30	251·2	882	72	306·1	396
31	253·1	855	73	307·1	391
32	255·0	831	74	308·0	386
33	256·8	808	75	308·9	381
34	258·6	786	76	309·9	377
35	260·3	765	77	310·8	372
36	262·0	746	78	311·7	368
37	263·7	727	79	312·6	364
38	265·3	710	80	313·5	359
39	266·9	693	81	314·3	355
40	268·4	677	82	315·2	351
41	269·9	662	83	316·1	348
42	271·4	647	84	316·9	344
43	272·9	634	85	317·8	340
44	274·3	620	86	318·6	337
45	275·7	608	87	319·4	333
46	277·1	596	88	320·3	330
47	278·4	584	89	321·1	326
48	279·7	573	90	321·9	323
49	281·0	562	91	322·7	320
50	282·3	552	92	323·5	317
51	283·6	542	93	324·3	313
52	284·8	532	94	325·0	310
53	286·0	523	95	325·8	307
54	287·2	514	96	326·6	305
55	288·4	506	97	327·3	302
56	289·6	498	98	328·1	299
57	290·7	490	99	328·8	296
58	291·9	482	100	329·6	293
59	293·0	474	105	333·2	281
60	294·1	467	120	343·3	249
61	294·9	460	135	352·4	224
62	295·9	453	150	360·8	203
63	297·0	447	165	368·5	187
64	298·1	440	180	375·6	173
65	299·1	434	195	382·3	161
66	300·1	428	210	388.6	150
67	301·2	422	225	394·6	141
68	302·2	417	240	400·2	133

Examples on the Use of the foregoing Table.

Example 1.—Given the area of the piston of a high-pressure engine 400 square inches, the length of the stroke 4 feet, the evaporation of water in the boiler half a cubic foot per minute, the pressure of the steam in the cylinder 60lbs. per square inch; to find the useful load and the horse-powers of the engine.

By the rule, page 12,

$$\frac{(60 - 16) \times 7}{8} = 38{\cdot}5 \text{ lbs.} = \text{useful load.}$$

To a pressure of 60 lbs. the corresponding volume in the table is 467, which being multiplied by the evaporating power of the boiler gives $467 \times \frac{1}{2} = 233\frac{1}{2} =$ number of cubic feet evaporated per minute.

The number of cubic feet discharged per stroke is equal to the area of the piston in feet, multiplied by the length of the stroke in feet $= \frac{400}{144} \times 4 = \frac{1600}{144} = 11\frac{1}{9}$.

The whole volume discharged per minute is equal to the number of strokes per minute, multiplied by the volume discharged at one stroke; and the volume discharged per minute must be equal to the volume evaporated per minute.

Number of strokes per minute $\times\ 11\frac{1}{9} =$ whole discharge in one minute $= 233\frac{1}{2}$ lbs. evaporated also in one minute.

Hence the number of strokes

$$= \frac{233\frac{1}{2}}{11\frac{1}{9}} = 21.$$

But the useful work done in one stroke is

$$38{\cdot}5 \times 400 \times 4 = 61600;$$

therefore the useful work per minute is

$$61600 \times 21 = 1293600;$$

and hence the horse-powers

$$= \frac{1293600}{33000} = 39\frac{1}{5}.$$

Example 2.—In a condensing engine the area of the

cylinder is 1440 square inches; the length of the stroke, including clearance, is 5 feet; the steam is cut off at 1 foot of the stroke; the clearance is $\frac{1}{4}$ of a foot; the pressure of the steam is 30 lbs. per square inch; the elasticity of the vapour in the condenser is 4 lbs.; the effective evaporation of the boiler is $\frac{1}{5}$ of a cubic foot per minute, and the resistances as usual: required, the useful load and the horse-powers.—*Tate's Mechanics.*

The space through which the piston moves before the steam is cut off is $= 1 - \frac{1}{4} = \frac{3}{4}$, and the whole length of the stroke is $= 5 - \frac{1}{4} = 4\frac{3}{4}$.

Then $\log (4\frac{3}{4} \div \frac{3}{4}) = \log 6{\cdot}33 = 1{\cdot}8453$, and $1{\cdot}8453 + 1 = 2{\cdot}8453$.

Hence $30 \times \frac{3}{4} \times 2{\cdot}8453 = 64{\cdot}01925$ = the whole work done in one stroke.

$$\text{Mean pressure} = \frac{64{\cdot}01925}{4\frac{3}{4}} = 13{\cdot}48 \text{ lbs.}$$

By the rule, page 12, the useful load

$$= \frac{(13{\cdot}48 - 5) \times 7}{8} = 7{\cdot}42.$$

By the Table, one cubic foot of water expands into 882 cubic feet of steam, at a pressure of 30 lbs. per square inch.

Hence the volume of steam evaporated per minute is

$$\tfrac{1}{5} \times 882 = 176{\cdot}4 \text{ cubic feet.}$$

The volume of steam discharged in one stroke is

$$\frac{1440}{144} \times 1 = 10 \text{ cubic feet.}$$

Hence the number of strokes per minute is

$$\frac{176{\cdot}4}{10} = 17{\cdot}64.$$

Now, the useful work per minute is found by multiplying together the area of the piston, the useful load, the length of the stroke, and the number of strokes per minute; hence

$$\text{Horse powers} = \frac{1440 \times 7{\cdot}42 \times 4\frac{3}{4} \times 17{\cdot}64}{33000} = 27{\cdot}13.$$

THEORETICAL INVESTIGATION OF THE PROPERTIES OF STEAM.

Let s = number of feet described at any part of the stroke of the piston; p' = pressure when that part is described; q = number of feet described before the steam is cut off, and l = length of the stroke in feet: by Mariotte's law,

$$p' : p :: q : s,$$

$$\text{whence } p' = \frac{q\,p}{s}.$$

Hence the variable work $\int \frac{q\,p\,ds}{s}$ done by expansion, taken between proper limits, is

$$\int_q^l \frac{q\,p\,d\,s}{s} = q\,p\,(\log l - \log q) = q\,p \log \frac{l}{q}.*$$

Put $\lambda = \frac{l}{q}$; then the work done by expansion will be represented by $qp \log \lambda$; and the work done before expansion is evidently represented by qp.

Hence the whole work done per square inch, both before and after expansion, is

$$qp + qp \log \lambda = q\,p\,(1 + \log \lambda). \quad \text{(Rule 1, page 3).}$$

Let L = load per square inch; then $L\,l$ = work exerted upon the load by each stroke; and as the work of the steam must be equal to the work done upon the load, there results

$$L\,l = q\,p\,(1 + \log \lambda),$$

$$\therefore L = \frac{qp}{l}(1 + \log \lambda). \quad \text{(Rule 2, page 3).}$$

From the above formula there evidently results

$$p = \frac{L\,l}{q\,(1 + \log \lambda)}. \quad \text{(Rule 3, page 3.)}$$

To determine the velocity of the piston when any part of

* The logarithms here used are the hyperbolic.

the stroke is described, the principle of *vis viva** must be applied. Poncelet has demonstrated, in his *Mechanique Industrielle*, that the work accumulated in any body or machine is half the *vis viva*; so that if $U =$ the units of work done upon any load or body, the weight of which is W, and through any space s, $U' =$ the units of work expended upon the resistance which opposes the motion of the body through the same space; then $U - U' =$ the units of work accumulated in the body; now, if $V =$ velocity of the body, and $g =$ accelerative force of gravity, the *vis viva* is $\frac{W}{g} . V^2$; therefore the accumulated work is

$$\tfrac{1}{2} . \frac{W}{g} . V^2 = U - U'.$$

Assuming that the whole mass or body moves with the same velocity as the piston, the work done upon each square inch by the steam, when s feet of the stroke have been described, is

$$q p \left(1 + \log \frac{s}{q}\right);$$

and the work expended on the load up to that point is $L . s$; therefore the work accumulated upon each square inch of the piston will be expressed by

$$q p \left(1 + \log \frac{s}{q}\right) - L s;$$

or, if $r =$ radius of the piston, and $\pi = 3{\cdot}1416$, then

$$\pi r^2 \left\{ q p \left(1 + \log \frac{s}{q}\right) - L s \right\} = U - U',$$

which is the accumulated work on the whole piston;

$$\therefore \frac{W V^2}{2 g} = \pi r^2 \left\{ q p \left(1 + \log \frac{s}{q}\right) - L s \right\},$$

$$\text{and } V^2 = \frac{2 \pi g r^2}{W} \left\{ q p \left(1 + \log \frac{s}{q}\right) - L s \right\};$$

* The *vis viva* of a body is its mass multiplied by the square of its velocity.

whence, by substituting for L its value, $\frac{qp}{l}(1 + \log \lambda)$, there results

$$V^2 = \frac{2\pi g r^2}{W}\left\{qp\left(1 + \log \frac{s}{q}\right) - \frac{qps}{l}(1 + \log \lambda)\right\}$$

$$= \frac{2\pi g q p r^2}{W}\left\{1 + \log \frac{s}{q} - \frac{s}{l}(1 + \log \lambda)\right\}.$$

To find when the velocity of the piston is a maximum, the part of the above formula within the vinculum must be a maximum, that is,

$$1 + \log \frac{s}{q} - \frac{s}{l}(1 + \log \lambda) \text{ must be a maximum;}$$

$$\text{or, } 1 + \log s - \log q - \frac{s}{l}(1 + \log \lambda) = \text{maximum,}$$

which put $= u$; then by differentiation,

$$\frac{du}{ds} = \frac{1}{s} - \frac{1}{l}(1 + \log \lambda) = 0,$$

$$\text{whence } s = \frac{l}{1 + \log \lambda}.$$

The maximum is indicated by $\frac{d^2 u}{d s^2}$, being evidently negative; hence the value of s, being substituted in the formula for the velocity, gives

$$V^2 = \frac{2\pi g q p r^2}{W}\left\{1 + \log \frac{l}{q(1 + \log \lambda)} - \frac{1}{1 + \log \lambda}(1 + \log \lambda)\right\}$$

$$= \frac{2\pi g q p r^2}{W}\left\{\log \frac{l}{q(1 + \log \lambda)}\right\} \text{ for the greatest or}$$

maximum velocity.

This result can be obtained without the aid of the Differential Calculus. See Professor Hann's edition of "Tredgold on the Steam Engine;" see also his "Theory of the Steam Engine," page 98.

PAMBOUR'S THEORY OF THE STEAM ENGINE.

Pambour, in his excellent work on this subject, grounds his theory on the two following principles:—

First, *that the engine having attained uniform motion, there is necessarily an equilibrium between the pressure of the steam in the cylinder, and the resistance against the piston*, that is

$$p = \mathrm{R} \text{.......... ....} \quad (1)$$

R being the whole resistance against the piston.

Secondly, *that there is necessarily an equality between the production of steam and its expenditure.*

Let P = pressure of steam in the boiler, A = area of the piston, v = velocity of the piston in a unit of time, S = volume of water evaporated in the same unit of time, m = ratio of the volume of steam formed under the pressure P of the boiler to the volume of water that has produced it; then m S will be the volume of steam formed in the same unit of time under the pressure P; and assuming, according to Mariotte's law, that the temperature remains the same, the volume m S of steam supplied each unit of time by the boiler when transmitted to the cylinder, will become $m\,\mathrm{S} \,.\, \frac{\mathrm{P}}{p}$; and A v being the expenditure of steam, the second principle can be expressed as follows:—

$$\mathrm{A}\,v = m\,\mathrm{S} \,.\, \frac{\mathrm{P}}{p} \text{..................} \quad (2)$$

But by equation (1) $p = \mathrm{R}$,

$$\therefore \mathrm{A}\,v = m\,\mathrm{S} \,.\, \frac{\mathrm{P}}{\mathrm{R}} \text{..............} \quad (3)$$

If the unit of time be one minute, and N the number of strokes in that time, l being the length of the stroke, and q the space described before the steam is cut off; then

$$v = \mathrm{N}\,l, \text{ or } \mathrm{N} = \frac{v}{l};$$

and since the expenditure of steam per minute is

$$\mathrm{A}\,q\,\mathrm{N},$$

there evidently results $\frac{A q v}{l}$ = expenditure of steam.

Hence, by substitution, equation (2) becomes

$$\frac{A q v}{l} = m S . \frac{P}{p}$$

$$\therefore p = \frac{m S P l}{A q v}.$$

Now, the whole work done per square inch, both before and after expansion, is

$$q p \,(1 + \log \lambda), \lambda \text{ being } = \frac{l}{q}, \text{ as before;}$$

and the pressure of the steam in the cylinder is evidently

$$A q p \,(1 + \log \lambda);$$

and, if R be the resistance per square inch against the piston, the whole resistance will be expressed by

$$R . A l;$$

hence, by the first principle, there results

$$A q p \,(1 + \log \lambda) = R . A l,$$

$$\text{or, } q p \,(1 + \log \lambda) = R l \;\ldots\ldots\ldots\; (4)$$

$$\text{but } p = \frac{m S P l}{A q v}$$

$$\therefore \frac{m S P l}{A v} (1 + \log \lambda) = R l;$$

$$\text{hence } v = \frac{m S P}{A R} (1 + \log \lambda) \ldots (5)$$

This is the same formula as that deduced by Pambour, in a different manner, at page 107 of his work, when the clearance is neglected.

But when the clearance is taken into account, the work done per square inch, both before and after expansion, becomes

$$p\,(q + c) \log \left(\frac{l + c}{q + c}\right) + q p$$

hence the first relation becomes

$$A \left\{ p\,(q + c) \log \left(\frac{l + c}{q + c} \right) + qp \right\} = \mathrm{R}\,A\,l,$$

$$\text{or } p\,(q + c) \left\{ \frac{q}{q + c} + \log \left(\frac{l + c}{q + c} \right) \right\} = \mathrm{R}\,l.$$

For the second relation there will result

$$\frac{A\,v\,(q + c)}{l} = m\,\mathrm{S}\,.\,\frac{\mathrm{P}}{p},$$

$$\therefore v = \frac{m\,\mathrm{S}\,\mathrm{P}}{A\,\mathrm{R}} \left\{ \frac{q}{q + c} + \log \left(\frac{l + c}{q + c} \right) \right\} \text{......} \quad (6)$$

c being the clearance.

The relation between the pressure and temperature of steam is of such great importance that numerous experiments have been made, both in this country and in France, to ascertain the pressure when the temperature is known, or to determine the temperature when the pressure is known. These important subjects will be forthwith detailed.

Arago and Dulong made the most extensive and careful experiments ever yet undertaken on this important subject, at the expense of the French government; these experiments range from 15 lbs. to 360 lbs. per square inch of pressure, or from 1 to 24 atmospheres; and the following formula has been the result of the researches of these eminent philosophers, which will represent temperature as compared with the pressure, without any sensible error,

$$p = (\cdot 26973 + \cdot 0068031\, t)^5 \text{.........} \quad (7)$$

$$\text{whence } t = 146\cdot 96 \sqrt[5]{p} - 39\cdot 644;$$

in which p is the pressure in pounds per square inch, and t the temperature in degrees of Fahrenheit.

Tredgold's formula for pressures from 1 to 4 atmospheres is sufficiently accurate, and is as follows,

$$p = \left(\frac{103 + t}{201\cdot 18} \right)^6 \text{..................} \quad (8)$$

$$\text{whence } t = 201\cdot 18 \sqrt[6]{p} - 103.$$

Pambour's formula for the same range is the following,

$$p = \left(\frac{98{\cdot}8 + t}{198{\cdot}56}\right)^6 \dots\dots\dots\dots\dots\dots \quad (9)$$

$$\text{whence } t = 198{\cdot}56\sqrt[6]{p} - 98{\cdot}8.$$

An important property of elastic fluids has been discovered by the famous Gay-Lussac, *i. e.*, that if the temperature of a given weight of any elastic fluid be made to vary, it will acquire augmentations of volume exactly proportional to the augmentations of temperature, and for every increase of one degree of temperature of Fahrenheit's thermometer there will be produced an increase of ·00202 of the volume of the fluid from the temperature of 32°.

If v be the volume of any given weight of elastic fluid under any pressure, and at 32° of Fahrenheit's scale, the volume v_1 which it will occupy under the same pressure, and at any other temperature t of Fahrenheit, will be

$$v_1 = v + v \times {\cdot}00202\,(t - 32).$$

This will also hold if we put the ratio of the relative volumes u and u_1, instead of the ratio of the absolute volumes v and v_1; there will thence result,

$$\frac{u}{u_1} = \frac{1 + {\cdot}00202\,(t - 32)}{1 + {\cdot}00202\,(t' - 32)}.$$

Neither the law of Gay-Lussac nor that of Mariotte will apply to the variations which take place in the steam while in contact with the water from which it is evaporated; but from the two laws a third law may be obtained, by which the variations of the volume of steam, when there is a change in both the temperature and pressure at the same time, may be determined. When it is required to find the volume of a given weight of steam, which changes from the temperature t' and the pressure p' to the temperature t and the pressure p, it may be assumed that the steam passes first from the pressure p' to the pressure p without changing its temperature, then by Mariotte's law there will result,

$$u_{\prime\prime} = u_{\prime}\,\frac{p'}{p}.$$

Again, assume that the steam passes from the temperature

t' to the temperature t without any change of pressure; then, by the law of Gay-Lussac, there results,

$$u = \left\{\frac{1 + \cdot 00202\,(t - 32)}{1 + \cdot 00202\,(t' - 32)}\right. u_{\prime\prime} = \frac{p'}{p}\left\{\frac{1 + \cdot 00202\,(t - 32)}{1 + \cdot 00202\,(t' - 32)}\right. u_{\prime}.$$

As this formula gives the law of the relative volumes, when both temperature and pressure change at the same time, there must be substituted in the above formula the values of p' and t', the pressure and temperature corresponding to each other in steam in contact with water, and there will result the required relative volumes. It is well known that under the pressure of the atmosphere, which is 14·7 lbs. per square inch, and at the temperature of 212° of Fahrenheit, the relative volume of steam in contact with the water from which it has been generated, is 1700 times that of the water, whence

$$u = 1700 \times \frac{14 \cdot 7}{p} \times \frac{1 + \cdot 00202\,(t - 32)}{1 + \cdot 00202\,(t' - 32)}.$$

By this equation, and any one of the equations, Arts. (7), (8), and (9), t may be eliminated. It will be preferable to use Tredgold's formula for pressures from 1 to 4 atmospheres, and from 4 and upwards, that of Dulong and Arago.

It will now be proper to give the formulæ, showing the relation between the pressure and volume, derived from experiment, where the principle of expansion is used in the steam engine.

Pole gives the following formula,

$$P = \frac{24250}{V - 65} \dots\dots\dots\dots\dots (a)$$

$$\text{whence} \quad V = \frac{24250}{P} + 65.$$

Here P is the pressure in pounds per square inch, and V its relative volume compared with that of its constituent water.

The accuracy of this formula may be relied upon throughout the range generally required in the Cornish engines, *i.e.*, from 65 lbs. to 5 lbs.

Pambour gives for condensing engines,

$$u = \frac{10000}{\cdot 4227 + \cdot 00258\,p}; \dots\dots\dots (b)$$

and for non-condensing engines,

$$u = \frac{10000}{1{\cdot}421 + {\cdot}0023\,p}; \;\ldots\ldots\ldots\ldots\; (c)$$

in which u is the volume and p the pressure in pounds per square foot.

Navier gives the following formula,

$$u = \frac{1000}{{\cdot}09 + {\cdot}0000484\,p}; \ldots\ldots\ldots\ldots\; (d)$$

in which u is the volume corresponding to the pressure p expressed in kilogrammes per square metre.

Let it be assumed that a volume of water E is converted into steam at a pressure p, and that M is the volume of steam which can be produced by it; then, by giving the preceding formulæ more general application, we may take

$$u = \frac{M}{E} = \frac{1}{\alpha + \beta p} \quad \ldots\ldots\ldots\ldots\ldots\; (e)$$

Now, if the same volume of water be converted into steam at the pressure $p_{\prime}$, and $M_{\prime}$ be the corresponding volume of steam, there will result $\frac{M_{\prime}}{E} = \frac{1}{\alpha + \beta p}$; whence, between the absolute volumes of steam, which correspond to the same weight E of water, there will result. by eliminating E by means of formula (e), the following relation,

$$\frac{M}{M_{\prime}} = \frac{\frac{\alpha}{\beta} + p_{\prime}}{\frac{\alpha}{\beta} + p},$$

$$\therefore p = \frac{M_{\prime}}{M}\left(\frac{\alpha}{\beta} + p_{\prime}\right) - \frac{\alpha}{\beta}, \;\ldots\ldots\ldots\; (f)$$

in which α and β are the constants in Pambour's formula.

By proceeding in the same manner with Pole's formula there results,

$$\frac{M_{\prime}}{M} = \frac{\frac{\alpha}{P_{\prime}} + \beta}{\frac{\alpha}{P} + \beta}, \;\ldots\ldots\ldots\ldots\ldots\ldots\; (g)$$

for the ratio of the volumes, where $\alpha = 24250$ and $\beta = 65$.

Pambour states that he has found, from a great number of experiments, that the steam, during its action in the cylinder, is always in the condition of maximum density for its temperature, and that when the pressure of the steam changes therein, its temperature changes at the same time, and that they always preserve that relation, which connects the pressures and temperatures of the steam in contact with the water from which it is generated.

ON THE WORK PERFORMED ON THE PISTON OF A STEAM ENGINE IN A GIVEN TIME.

Put A = area of the piston in square feet,
P = pressure in the boiler,
p = pressure at the xth foot of the stroke,
p' = pressure in the cylinder before expansion,
l = actual length of the stroke,
q = that part of the stroke before the steam is cut off,
c = the clearance,
E = number of cubic feet of water converted into steam per minute,
N = number of single strokes per minute,
and U = units of work of the steam per minute.

Since each cubic foot of water, converted into steam existing in the cylinder before expansion, begins under a pressure p', it therefore occupies a relative space by formula (e), represented by

$$u = \frac{1}{\alpha + \beta p'};$$

while the number of cubic feet of water which is evaporated in the boiler, and passes into the cylinder in the form of steam at every stroke of the piston, is represented by $\frac{E}{N}$; therefore the space occupied in cubic feet in the cylinder, when the valve is closed and expansion begins, will be represented by

$$\frac{E}{N}\left(\frac{1}{\alpha + \beta p'}\right).$$

In the same manner the space occupied at the xth foot

of the stroke, when the pressure p' becomes p, will be expressed by

$$\frac{\mathrm{E}}{\mathrm{N}}\left(\frac{1}{a + \beta\, p}\right).$$

But that space in the cylinder, filled with steam before expansion begins, is also represented by A $(q + c)$; and the space in the cylinder occupied at the xth foot of the stroke by A $(x + c)$. Hence there results the following equations,

$$\frac{\mathrm{E}}{\mathrm{N}}\left(\frac{1}{a + \beta\, p}\right) = \mathrm{A}\,(x + c),$$

$$\text{and} \quad \frac{\mathrm{E}}{\mathrm{N}}\left(\frac{1}{a + \beta\, p'}\right) = \mathrm{A}\,(q + c);$$

whence from the former equation,

$$\mathrm{N}\,(a + \beta\, p) = \frac{\mathrm{E}}{\mathrm{A}\,(x + c)} \quad \ldots\ldots (h)$$

and from the latter,

$$\mathrm{N} = \frac{\mathrm{E}}{\mathrm{A}\,(q + c)\,(a + \beta\, p')};$$

$$\therefore p = \frac{\mathrm{E}}{\mathrm{AN}} \cdot \frac{1}{\beta\,(x + c)} - \frac{a}{\beta} \quad \ldots\ldots\ldots (k)$$

But the work done by expansion per square foot upon the piston is expressed by $\int_q^l p\, d\, x$, and by substitution from (k) there results,

$$\int_q^l p\, d\, x = \frac{\mathrm{E}}{\mathrm{A\,N}\,\beta} \cdot \int_q^l \frac{d\, x}{x + c} - \int_q^l \frac{a\, d\, x}{\beta}$$

$$= \frac{\mathrm{E}}{\mathrm{A\,N}\,\beta} \cdot \log\left(\frac{l + c}{q + c}\right) - \frac{a}{\beta}\,(l - q);$$

which expresses the whole work done by expansion; and, since the pressure upon the piston, before expansion begins, is represented by p', and that it moves q feet under this

pressure, whence the work performed on each square foot is, by formula (k), represented by

$$q p = \frac{E}{A N \beta}\left(\frac{q}{q+c}\right) = \frac{a q}{\beta};$$

therefore the sum of these two values is the whole work done upon each square foot of the piston, both before and after expansion, which sum is

$$\frac{E}{A N \beta}\left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}\right\} - \frac{a}{\beta} l;$$

which, being multiplied by A N, will give the whole work done on the piston per minute, or

$$U = \frac{E}{\beta}\left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}\right\} - \frac{a A N l}{\beta} \ldots (m)$$

Now, the work produced by the power per minute must be equal to the resistance, when the engine has attained uniform motion, and R being the whole resistance, there results

$$\frac{E}{\beta}\left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}\right\} - \frac{a}{\beta} \cdot A N l = A N R l,$$

$$\text{and } A N l\left(R + \frac{a}{\beta}\right) = \frac{E}{\beta}\left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}\right\}.$$

By substituting v, in the last formula, for its equal N l, which represents the velocity of the piston per minute, there results

$$A v\left(R + \frac{a}{\beta}\right) = \frac{E}{\beta}\left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}\right\},$$

$$\text{whence } v = \frac{E}{A(a + \beta R)} \cdot \left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}\right\} \ldots (n)$$

The quantity R, being the whole resistance acting on a unit of surface of the piston, comprehends the resistance arising from the useful load, and from friction.

Let the resistance from the motion of the useful load $= \rho$, the friction of the unloaded engine $= f$, the quantity the friction is augmented by each unit of the useful load $= \delta$, and the resistance from imperfect condensation $= h$; then

$$Z = \rho(1 + \delta) + f + h.$$

By substituting this value of R in formula (n) there results

$$v = \frac{E}{A} \cdot \frac{1}{a + \beta\{(1+\delta)\rho + f + h\}} \cdot \left\{ \log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c} \right\} \ldots (o)$$

When the engine does not work expansively; then $q = l$, and

$$v = \frac{E}{A} \cdot \frac{1}{a + \beta\{(1+\delta)\rho + f + h\}} \cdot \left(\frac{l}{q+c}\right) \ldots\ldots (p)$$

since $\log\left(\frac{l+c}{q+c}\right)$ becomes $\log\left(\frac{l+c}{l+c}\right) = \log 1 = 0.$

One of Pambour's fundamental formula is (o), by means of which he determines the load for a given velocity, by finding ρ from it, which is the resistance for a unit of surface of the piston; and hence the load on the whole piston is $A\rho$, and

$$A\rho = \frac{E\left\{ \log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c} \right\}}{(1+\delta)\beta v} - \frac{A}{1+\delta} \cdot \left(\frac{a}{\beta} + h + f\right) \ldots (q),$$

and from formula (o), the evaporation of an engine to give motion to a load ρ, at the velocity v, is

$$E = A v \cdot \frac{a + \beta\{(1+\delta)\rho + h + f\}}{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c}} \ldots\ldots\ldots (r)$$

From formula (q), the useful effect which an engine can produce in a unit of time, at the velocity v, is

$$A\rho v = \frac{E\left\{ \log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c} \right\}}{(1+\delta)\beta} - \frac{A v}{1+\delta} \cdot \left(\frac{a}{\beta} + h + f\right) \quad (s)$$

The useful effect in terms of the load is found by multiplying formula (o) by $A\rho$, and there results

$$A\rho v = \frac{E\rho\left\{ \log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c} \right\}}{a + \beta(1+\delta)\rho + h + f} \ldots\ldots\ldots\ldots (t)$$

which, being divided by 33,000, will give the horse-powers of the engine.

Pambour takes $h = 4$ lbs. per square inch, and $f = \cdot 5$ lbs., consequently $h + f = 4{\cdot}5$ lbs. per square inch, or $144 \times 4{\cdot}5 = 648$ lbs. per square foot, $c = {\cdot}05\, l$, and $1 + \delta = 1{\cdot}14$ lbs.

ON THE DUTY OF THE STEAM ENGINE.

The *duty* of an engine is the amount of work yielded by that engine from the combustion of a given quantity of coals, as a bushel or cwt.; and is, therefore, a measure of cost with respect to fuel.

Mr. Pole, who is the highest authority in this country on these subjects, when speaking of the great duty of the Cornish engines, observes, "Another cause of the great duty is the *use of high-pressure steam.* This is not only advantageous indirectly, inasmuch as it enables the expansive principle to be applied with greater effect, but there is a totally distinct economical advantage in the use of high-pressure steam *per se,* which is often overlooked, and deserves special mention. It is founded on the principle, *that the pressure of the steam* increases in a greater ratio than its density; whence it follows that the higher the pressure the steam is raised to, the less *proportionate* quantity of water it contains, and therefore the less fuel is consumed, since a given quantity of fuel will evaporate the same weight of water at all temperatures."

Mr. Pole gives, at page 169 of his work on the Cornish Engine, the following example:—

Example.—A Cornish engine, with a cylinder 70 inches in diameter, and a 10-feet stroke, has the steam admitted at a pressure of 45 lbs. to the square inch (*i.e.*, 30 lbs. above the atmospheric pressure) during $\frac{1}{6}$ of the stroke, and during the remainder the steam is allowed to expand; required the duty of the engine.

The diameter of the cylinder being 70 inches, the area is 3848 square inches, which multiplied by 45 lbs., the pressure on each square inch, give the whole pressure on the piston $= 173160$ lbs. This weight is moved through $1\frac{2}{3}$ feet, the space during which the steam is admitted; hence a quantity of work $= 173160 \times 1\frac{2}{3} = 288600$ lbs. raised one foot high before the steam is cut off.

Hence the whole work on the piston, both before and after the expansion, by Rule 1, is

$$288600 \, (1 + \log 6) = 288600 \times 2{\cdot}79176 = 805702 \text{ lbs.}$$

Since the area of the cylinder is 3848 square inches, and the steam is admitted during $1\frac{2}{3}$ feet $= 20$ inches of the stroke, the quantity of fuel consumed to do the above work

$= 20 \times 3848 = 76960$ cubic inches; and since the relative densities of water and steam at 45 lbs. pressure are as 1 to 608 (see table, p. 13), this volume of steam will contain $\frac{76960}{608}$ $= 126$ cubic inches of water; and as each cubic foot of water $= 1728$ cubic inches, weighs 62·5 lbs., we shall have

$$1728 : 126 :: 62{\cdot}5 : 4{\cdot}55,$$

that is, 126 cubic inches of water weighs 4·55 lbs.

Now it has been found by experiment that 1 lb. of coal will evaporate 9·27 lbs. of water; hence the quantity of coal required for each stroke $= \frac{4{\cdot}55}{9{\cdot}27} = {\cdot}49$ lbs. = nearly ½ lb. of coal, which is required for each stroke, to produce a motive power of 805702 lbs. raised one foot high. At this rate, one bushel = 94 lbs. of coal will give the following amount,

$$\frac{805702 \times 94}{{\cdot}49} = 154{,}563{,}445 \text{ lbs.},$$

which is a very large duty; but this duty will be somewhat reduced by taking in the minutiæ given in formula (t).

If B = number of bushels of coals consumed in an engine in any given time $= t$ minutes, D = duty of the engine, and U = units of work per minute; then $U\,t$ = units of work yielded by the engine while B bushels of coals are being consumed; hence $\frac{U\,t}{B}$ will be the units of work yielded by each bushel of coals consumed by that engine; therefore,

$$D = \frac{U\,t}{B}.$$

To find the Point at which the Steam must be cut off to obtain the greatest quantity of Useful Work.

By multiplying formula (h) by l there results

$$N\,l = \frac{E\,l}{A\,(q + c)\,(\alpha + \beta p')},$$

and, since $N\,l = v$, by substitution,

$$v = \frac{E\,l}{A\,(q + c)\,(\alpha + \beta p')},$$

in which, by putting P the pressure in the boiler for p', there will result the *velocity v' for the maximum useful effect,*

$$v' = \frac{E\,l}{A\,(q+c)\,(a+\beta P)};$$

which, being substituted in formula (s), gives

$$A\rho v' = \frac{E}{1+\delta)\beta}\left\{\log\left(\frac{l+c}{q+c}\right) + \frac{q}{q+c} - \frac{l}{q+c}\cdot\frac{a+\beta(h+f)}{a+\beta P}\right\}.$$

By differentiating this formula for q, the part of the stroke where the steam should be cut off so as to give the useful effect a maximum, or the greatest possible, there results

$$\frac{q}{l} = \frac{a+\beta(h+f)}{a+\beta P},$$

or

$$\frac{q}{l} = \frac{1}{\frac{a+\beta P}{a+\beta(h+f)}},$$

that is, in other words,

$$\frac{q}{l} = \frac{\textit{volume at pressure } P}{\textit{volume at pressure } p'}.$$

By formula (u), given by Pole, this will become

$$\frac{q}{l} = \frac{\frac{a}{P}+}{\frac{a}{h+f}+\beta}, \quad \text{or } q = l.\frac{\frac{a}{P}+\beta}{\frac{a}{h+f}+\beta},$$

in which $a = 24250$, and $\beta = 65$.

Example 1.—Let the length of the stroke of the piston be 10 feet, the pressure of the steam 40 lbs. per square inch, and the useless resistance $h+f = 5$ lbs.; it is required to find at what part of the stroke the steam must be cut off to give the greatest amount of useful work.

By the preceding formula

$$q = 10\cdot\frac{\frac{24250}{40}+65}{\frac{24250}{5}+65} = 1{\cdot}36 \text{ feet},$$

the required part of the stroke where the steam must be cut off to give the greatest amount of useful work.

Example 2.—The length of the stroke, including clearance, which is $\frac{1}{2}$ foot, is 10·5 feet, the pressure of the steam in the cylinder is 50 lbs. per square inch, the elasticity of vapour in the condenser is 4 lbs., and the whole resistance from friction $2\frac{1}{2}$ lbs. per square inch; required the point at which the steam must be cut off to produce the maximum or greatest effect.

By Pole's formula,

$$q = \frac{\frac{a}{P} + \beta}{\frac{a}{h+f} + \beta} \cdot l = \frac{\frac{24250}{50} + 65}{\frac{24250}{6 \cdot 5} + 65} = \frac{550 \times 10 \cdot 5}{3795 \cdot 7} = 1 \cdot 52,$$

which is the part of the stroke where the steam must be cut off, including clearance.

Example 3.—Let the surface of the piston A be 1800 square inches, the effective evaporation E be ·927 cubic feet per minute, the length of the stroke before the steam is cut off q be $1\frac{1}{2}$ feet, the other data being as in Example 2; required the number of strokes per minute.

$$\frac{A\,N\,q}{144} = E\left(\frac{24250}{50} + 65\right),$$

$$\therefore N = \frac{144 \times 550 \times \cdot 927}{1800 \times 1 \cdot 5} = 27,$$

which is the number of strokes of the piston to produce the maximum, or greatest effect.

Having now given the mathematical theory of the steam engine, with a variety of examples of its use, it will now be proper to advert to the different appendages of that splendid power.

ON THE SAFETY VALVE OF THE STATIONARY ENGINE.

In the appendix to "Tredgold on the Steam Engine," the following rule is given to find the area of the aperture for the safety valve:—

Divide the area of the fire-surface by the excess of pressure in the boiler above that of the atmosphere in pounds per

square inch, and the quotient will be the square of the diameter of the narrowest part of the aperture in inches.

The safety valve is usually loaded by means of a lever, with a weight to move along it to suit the required pressure.

To find the Pressure on the Valve to each Square Inch, when the whole Weight is put upon the Valve.

Multiply the square of the diameter of the valve by ·7854, and the product will give the area, or number of square inches in the valve; then divide the whole weight upon the valve in pounds by the number of square inches in the valve, and the quotient will give the number of pounds of pressure to each square inch in the valve.

Example.—If a weight of 112 lbs. be placed upon the valve, the diameter of which is 3 inches, required the pressure to each square inch.

$3^2 \times \cdot 7854 = 7$ square inches nearly; then $112 \div 7 = 16$ lbs., the required pressure per square inch.

ON THE LEVER OF THE SAFETY VALVE.

In addition to the weight moving along the lever, the weight of the lever must also be taken into the calculation; for when the lever is large, and the aperture of the valve small, the weight of the lever is such as to produce a very sensible pressure upon the valve.

The calculation for graduating the lever is as follows:—Divide the length by the distance between the fulcrum and valve, and the quotient is the leverage, which, multiplied by the weight, gives the whole weight on the valve; and this product, divided by the number of square inches in the valve, gives the pressure per square inch in the boiler.

Or, if the weight per square inch be known, multiply that weight by the number of square inches in the aperture of the valve, and this product gives the whole weight upon the valve, which, divided by the leverage, gives the weight which must be put on the end of the lever.

Example.—Given the whole length of the lever 32 inches, the distance between the fulcrum and valve 4 inches, the diameter of the valve $2\frac{1}{2}$ inches; required the weight to be put on at the end of the lever, so as to have a pressure of 50 lbs. per square inch upon the valve; also to divide the lever so as

to have 40, 30, 20 lbs., &c., upon the valve with the same weight.

$32 \div 4 = 8 =$ leverage,

$(2 \cdot 5)^2 \times \cdot 7854 = 4 \cdot 9 =$ area of the valve,

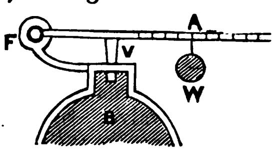
$4 \cdot 9 \times 50 = 245$ lbs. $=$ whole weight on the valve,

$\frac{245}{8} = 30\frac{5}{8}$ lbs. $=$ weight which must be put on at the end of the lever to give 50 lbs. per square inch,

and $\frac{4 \cdot 9 \times 40}{30\frac{5}{8}} = 6 \cdot 4$; then $6 \cdot 4 \times 4 = 25 \cdot 6$ inches $=$ distance from the fulcrum at which the weight must be put to have a pressure on the valve of 40 lbs. And $32 - 25 \cdot 6 = 6 \cdot 4$ inches, the distance which the weight must be moved towards the fulcrum to have 40 lbs. pressure per square inch; and for 30 lbs. per square inch it must be moved 6·4 inches more, &c.

To graduate the Lever of the Safety Valve, the Weight of the Lever being considered.

Let A F be a graduated lever, turning on F as fulcrum; V the valve, which is raised when the elastic force of the steam becomes too great for the pressure of the weight W, B being a portion of the boiler.



Put A F $=$ L, V F $= l$, W $=$ weight at A, $\omega =$ weight of the lever A F, $r =$ radius of the valve, and P $=$ greatest pressure of the steam in the boiler. Then $\pi r^2 =$ area of the aperture of the valve, πr^2 P $=$ pressure on the valve, and by the property of the lever,

$$L \times W + \tfrac{1}{2} L \times \omega = \pi r^2 P \times l, \text{ whence}$$

$$W = \frac{\pi r^2 l P - \frac{1}{2} L \omega}{L} \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

The weight W at the end of the lever may be determined from formula (1); after which the length L′, corresponding to any other pressure P′, may be found from the following formula, which is derived by substituting L′ for L, and P′ for P, in formula (1), and transposing, which gives

$$L' = \frac{\pi r^2 l P' - \frac{1}{2} L \omega}{W} \;\; \ldots\ldots\ldots\ldots \;\; (2)$$

Example 1.—Required the weight W, when A F = 24, V F = 3 inches, weight of lever = 4 lbs., radius of the aperture of the valve = $1\frac{1}{2}$ inches, and the pressure P of the steam in the boiler = 40 lbs. per square inch.

Here the area of the valve $\pi r^2 = 3\cdot1416 \times (1\frac{1}{2})^2 = 7\cdot07$ square inches nearly, which, by omitting the small decimal, may be taken as 7 square inches; whence

$$W = \frac{\pi r^2 l P - \frac{1}{2} L \omega}{L} = \frac{7 \times 3 \times 40}{24} = 35 \text{ lbs.}$$

That is, 35 lbs. put at the end of the lever will give the required pressure. We have next to find the distance L′ from F, at which the weight must be put on to give any other required pressure P′.

Example 2.—Let the pressure P′ be 30 lbs. per square inch, all the other dimensions and weights being the same as in the last example; required the distance L′.

By formula (2),

$$L' = \frac{\pi r^2 l P' - \frac{1}{2} L \omega}{W} = \frac{7 \times 3 \times 30 - 12 \times 4}{35} = 16\tfrac{22}{35} \text{inches,}$$

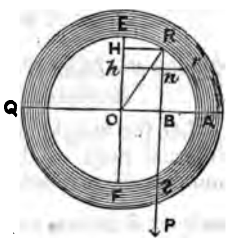
and $24 - 16\frac{22}{35} = 7\frac{13}{35}$ inches, which the weight W must be moved on towards F to give a pressure of 30 lbs. per square inch on the valve. Similarly, the distance on the lever may be found for a pressure of 20 lbs. to be $10\frac{22}{35}$ inches, or the weight W must be suspended at $24 - 10\frac{22}{35} = 13\frac{13}{35}$ inches from F, or $6\frac{24}{35}$ inches nearer to F than in the latter case, and a further distance of $6\frac{24}{35}$ towards F, to give a pressure of 10 lbs. per square inch: thus the graduation of the lever may be completed for any given pressure on the safety valve; for instance, if the intermediate pressures of 25 lbs., 15 lbs., &c., be required, the distance $6\frac{24}{35}$ inches must everywhere be divided into two equal parts, that is, $6\frac{24}{35} \div 2 = 3\frac{12}{35}$ inches, which will give the places where the weight must be suspended for these pressures.

THE FLY WHEEL.

When a moving power is supplied irregularly, as by the piston of a steam engine, the action of which is intermitting, or by impulses,—while numerous machines, with the exception of the pumping engines and a few others, require a regular force,—the method of regulating the motion of such machines is by means of a fly wheel, which is usually fixed on the crank-shaft of a stationary engine. The rim of this wheel consists of a ponderous mass of metal, revolving freely on its axis, and by its inertia produces a reservoir as well as a regulator of force; since a small surplus of force, acting for a short time, will accumulate a considerable power in the fly wheel; and this power, being applied suddenly for a short time, is capable of supplying the short intermissions of the moving power, and producing a near approximation to perfect regularity in the motion of the machinery. This method of regulating motion is not, for obvious reasons, applied to marine and locomotive engines.*

When the Force acts in one Direction only, to find the Limits within which the Angular Velocity of the Fly Wheel varies.

Let A R Q S be the fly wheel, O its centre, O R a crank, on which the rod P R acts in directions parallel to the diameter E F, and let P be a constant force acting on the rod P R; also let Q be a weight equivalent to the resistance of the machine, and acting perpendicularly at Q the extremity of the radius O Q. Draw R H, $r h$ perpendicular to E F, and indefinitely near to each other; also put $RO = r$, $QO = R$, and $\pi =$ circumference to radius $= 1$. Now let the point R move through the



indefinitely small space R r; then the force of P is measured by the product of the resolved part R n of the force P and the small space R r, which product is evidently $= P \times Rr = P \times Hh$, since the force nr is wholly ineffectual. Let the force P act from E to F, then H h becomes = the diameter E F, and the whole force from E to F will be $= P \times EF = P \times 2r$; and the whole dynamical effect of the resistance

* *See pages* 162, 163 of "A Treatise on Marine Engines," Weale's Ser

of the machine in one entire revolution, which is represented by Q, will be $= Q \times 2\pi R$; but since the whole effect of the force P is consumed by the useful and useless resistance of the machinery taken together, there will result

$$2 P r = 2 Q \pi R,$$

or, $$P = \frac{\pi Q R}{r}.$$

Now, let the resistance Q just balance the force P, when the crank is in the two positions O R, O S; and put the angle R O A $= \frac{1}{2}$ the angle R O S $= \phi$; then

$$P r \cos\phi = Q R;$$

and by substituting the value of P in this equation, there will result, after reduction,

$$\cos\phi = \frac{1}{\pi} = \frac{1}{3{\cdot}1416} = {\cdot}3183;$$

$$\therefore \phi = 71^\circ\ 26',$$

and the arc $R A S = 2\phi = 142^\circ\ 52'$.

Corollary.—For the double-acting engine we find in a similar manner,

$$\cos\phi = \frac{2}{\pi} = \frac{2}{3{\cdot}1416} = {\cdot}6366;$$

$$\therefore \text{arc } R S = 2\phi = 100^\circ\ 54'.$$

To find the Units of Work in the Fly Wheel, and the number of Revolutions which it will make before it stops.

Let the weight of the wheel $= W$ lbs., the external and internal radii of the rim respectively $= R$ and r feet, its number of revolutions per second $= s$, the diameter of its axis $= d$ inches, and the friction upon it $= \frac{1}{n}$ of the whole weight of the wheel, the inertia of the axle and spokes of the wheel being neglected as not materially affecting the result. (See last figure.)

Then the radius of gyration $k = \sqrt{\frac{R^2 + r^2}{2}}$. (See Baker's "Statics and Dynamics," Weale's Series.)

Units of work in the wheel

$$= s\left(2\pi\sqrt{\frac{R^2 + r^2}{2}}\right)^2 \times \frac{W}{2g}$$

$$= \frac{\pi^2 s^2 W}{g} (R^2 + r^2),$$

circumference of axis $= \frac{\pi d}{12}$,

work destroyed by friction in one revolution

$$= \frac{\pi d}{12} \times \frac{W}{n} = \frac{\pi g W}{12 n}.$$

Now, put N = number of revolutions made by the wheel before it stops; then

whole work destroyed by friction $= \frac{\pi d W}{12 n} \times N$,

$$\therefore \frac{\pi d W}{12 n} \times N = \frac{\pi^2 s^2 W (R^2 + r^2)}{g};$$

whence $$N = \frac{12 n \pi s^2 (R^2 + r^2)}{g d}.$$

Note.—This result is independent of the weight of the wheel.

Example.—The external and internal radii of the rim of a fly wheel are 5 and 3 feet, it makes 3 revolutions per second, the diameter of the axle is 2 inches, and the friction upon it $\frac{1}{10}$ of the weight of the wheel, or $n = 10$; how many revolutions will it make before it stops?

By the preceding formula, the number of revolutions, or

$$N = \frac{12 \times 10 \times 3{\cdot}1416 \times 3^2 (5^2 + 3^2)}{2 \times 32\frac{1}{6}} = 1720.$$

Practical Rules and Examples for the Fly Wheel of the Double-acting Engine.

The mean radius, the horse-powers, and the number of revolutions, or the number of double strokes per minute being given, *to find the weight of the wheel.*

Rule 1.—Multiply the number of horse-powers of the engine by 2275, and again by the denominator of the fraction

showing the variation from the mean velocity. Multiply the square of the mean radius by the cube of the number of revolutions per minute. Divide the former product by the latter, and the quotient will be the weight of the fly wheel in tons. See Hann and Gener's "Treatise on the Steam Engine," formula (21), p. 136.

Example.—A double-acting engine makes 15 revolutions per minute, the radius of the fly wheel is 15 feet, the horse-powers are 60; what must be the weight of the fly wheel, when the variation is $\frac{1}{40}$ from the mean velocity?

By the rule,

$$\frac{2275 \times 60 \times 40}{15^2 \times 15^3} = \frac{5460000}{759375} = 7\cdot2 \text{ tons.}$$

RULE 2.—Multiply the number of horse-powers by the denominator of the fraction showing the variation from the mean velocity, divide the product by the area of the section of the rim of the fly wheel, and extract the cube root of the quotient.

Multiply the cube root just found by 12·17, and divide the product by the number of revolutions per minute, and the quotient will be the *mean radius.* See Hann and Gener's "Treatise," formula (22), p. 136.

Example.—A double-acting steam engine makes 10 revolutions per minute, the horse-powers are 60, the area of the section of the rim is 1·3 square feet, and the variation $\frac{1}{40}$ from the mean velocity; required the mean radius of the fly wheel.

By the rule,

$$\sqrt[3]{\frac{60 \times 40}{1\cdot3}} = \sqrt[3]{1846\cdot15} = 12\cdot27,$$

and $\dfrac{12\cdot27 \times 12\cdot17}{10} = 14\cdot93$ feet $=$ mean radius of the wheel.

RULE 3.—Multiply 1803 by the number of horse-powers, and again by the denominator of the fraction showing the variation from the mean velocity, and divide this product by the product of the cubes of the mean radius and of the number of revolutions per minute, and the quotient will be the ***area of the section of the rim.***

Example.—A double-acting engine of 60 horse-powers makes 10 revolutions per minute, the mean radius of the

wheel is 12 feet, and the variation from the mean velocity is $\frac{1}{30}$; required the section of the rim of the fly wheel.

By the rule,

$$\frac{1803 \times 60 \times 20}{12^3 \times 10^3} = \frac{2163600}{1728000} = 1{\cdot}252 \text{ feet.}$$

Practical Rules and Examples for the Fly Wheel of the Single-acting Engine.

The number of single strokes per minute, the mean radius, and the horse-powers being given, *to find the weight of the wheel.*

RULE 1.—Multiply the number of horse-powers by 94880, and again by the denominator of the fraction showing the variation from the mean velocity, and divide this product by the product of the square of the mean radius and of the cube of the number of single strokes per minute, and the quotient will be the weight of the wheel in tons. See Hann and Gener "On the Steam Engine," formula (14), p. 133.

Example.—A single-acting engine of 20 horse-powers makes 30 single strokes per minute, the mean radius of the wheel is 13 feet, and the variation is $\frac{1}{25}$ from the mean velocity; required the weight of the fly wheel.

By the rule,

$$\frac{20 \times 94880 \times 25}{13^2 \times 30^3} = \frac{47440000}{4563000} = 10{\cdot}4 \text{ tons nearly.}$$

RULE 2.—Multiply the number of horse-powers by the denominator of the fraction showing the variation from the mean velocity, divide the product by the area of the section of the rim, and extract the cube root of the quotient; multiply the cube root, thus found, by 42·2, and divide the product by the number of single strokes per minute, and the quotient will be *the mean radius of the wheel.*

Example.—A single-acting engine of 60 horse-powers makes 40 single strokes per minute, the area of the section of the rim of the fly wheel is 1·2 square feet, and variation from the mean velocity $\frac{1}{80}$; required the mean radius of the wheel.

By the rule,

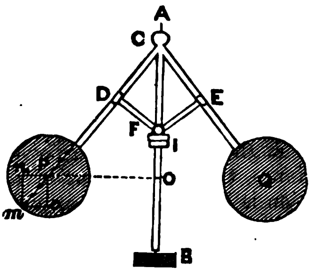
$$\sqrt[3]{\frac{60 \times 60}{12}} = \sqrt[3]{3000} = 14{\cdot}4,$$

and $\dfrac{42{\cdot}2 \times 14{\cdot}4}{40} = 15{\cdot}2$ feet nearly, the mean radius of the wheel.

THE GOVERNOR.

The governor is an apparatus for regulating the supply of steam from the boiler to the cylinder so as to give a constant and steady velocity to the engine.

A B is a vertical shaft turning freely on the sole B by its connection with the machinery of the steam engine; C P, C Q are two bars moving freely on the centre C, and carrying the two weights or balls P, Q; F D, F E are two rods connected to the bars at D and E, and attached to the collar I, which is capable of sliding freely up and down the shaft A B. This collar is united to a lever, which opens or closes the throttle valve, which supplies the cylinder with steam.



When A B revolves too fast, the balls, by their centrifugal force, fly outwards, raising the slide I, and partially closing the throttle valve; and when the shaft moves too slowly, the balls collapse, and the slide consequently descending, admits a more full supply of steam, thus regulating the engine to almost complete uniformity.

The ball or weight P is acted upon by two forces, *i.e.*, the centrifugal force in the direction P*n*, and gravity in the direction P*o*; to represent these two forces, complete the parallelogram P*nmo*; then we shall have the triangles P*mo*, A P O similar; and, if f represent the centrifugal force, and W the weight of the ball P, then,

$$\frac{f}{W} = \frac{PO}{CO}.$$

Also, by Art. 277, Baker's "Statics and Dynamics," Weale's Series,

$$f = \frac{W \,.\, V^2}{g \,.\, PO},$$

and by substituting the value of f in the preceding equation, there results after reduction,

$$CO = \frac{g \,.\, PO^2}{V^2},$$

or, if v = angular velocity per second of the governor, at a unit's distance from the shaft A B, then $V = PO \,.\, v$, and by substitution,

$$CO = \frac{g}{v^2}.$$

Now, if n = number of revolutions per minute, then $\frac{n}{60}$ = number per second,

$$\therefore v = \frac{2\pi n}{60} = \frac{\pi n}{30},$$

$$\text{whence} \qquad CO = \frac{30^2 \times g}{\pi^2 n^2} \text{ in feet,}$$

$$= \frac{30^2 \times 32\tfrac{1}{6} \times 12}{(3\cdot1416)^2 n^2} \text{ in inches,}$$

$$= \frac{35200}{n^2} \text{ inches nearly.}$$

Since the throttle valve of the steam engine cannot be opened without an adequate force exerted by the governor, which may be measured by finding what weight will produce that force: let p be the required weight, W being the weight of one the balls of the governor, as before; then Hann has shown in his works on the steam engine that

$$\frac{p}{W} = \frac{21}{100} \cdot \frac{CP}{CD},$$

and that, if $\frac{\text{OP}}{\text{OD}} = \frac{3}{2}$, which is the usual proportion in the governor,

$$\frac{p}{\text{W}} = \frac{21}{100} \cdot \frac{3}{2} = \frac{63}{200};$$

$$\therefore \text{W} = \frac{200\,p}{63} = 3{\cdot}174\,p;$$

and, if $p = 10$ lbs., then W $= 31\frac{3}{4}$ lbs. nearly, the required weight of one of the equal balls.

PRACTICAL RULES AND EXAMPLES ON THE GOVERNOR.

There are two conditions in calculations for the governor; which are, to find the position of the point F, attached to the collar I (see last fig.), when the governor performs the required number of revolutions; and to find what should be the range of the balls, or the radius of the circle described by them, that the supply of steam through the throttle valve may give the engine a steady speed.

To find the Distance of the Plane, in which the Balls revolve, from their Point of Suspension.

RULE.—Divide 35200 by the square of the number of revolutions per minute, and the quotient will be the distance required.

Example.—Required the distance of the plane, in which the balls of the governor revolve, from the point of suspension, when it makes 30 revolutions per minute.

By the rule,

$$\frac{35200}{30^2} = \frac{35200}{900} = 39{\cdot}11 \text{ inches.}$$

To find the Radius of the Circle described by the Balls.

RULE.—First, find the distance of the plane, in which the balls revolve, from their point of suspension, by the last rule. Secondly, from the square of the length of the arm subtract

the square of the distance last found, and the square root of the remainder will be the distance required.

Example.—The length of the arms of a governor is 30 inches, and it revolves 40 times per minute; required the radius of the circle described by the balls.

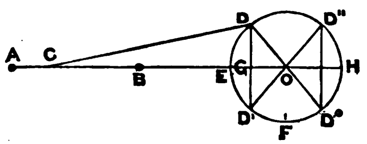
First, $\frac{35200}{40^2} = \frac{35200}{1600} = 22$ inches, the distance between the point of suspension and the plane of revolution.

Second, $\sqrt{30^2 - 22^2} = \sqrt{416} = 20{\cdot}4$ inches, the radius of the circle described by the balls.

PRINCIPLES OF THE CRANK.

The crank is used for converting rectilinear into rotatory motion.

To establish the principle that there is no loss of power in the use of the crank in steam engines, the celebrated mathematician, Mr. Woolhouse, has given, in his edition of "Tredgold's Steam Engine," the following dynamical investigation.



In the annexed figure let C D $= r$ be the length of the connecting rod, O D $= \rho$ the radius of the crank, the angle D O E $= a$, the velocity at C $= v$, the velocity at D $= v'$, and the distance A C $= x$; then the perpendicular D G $= \rho \sin^2 a$, C G $= \sqrt{r^2 - \rho^2 \sin^2 a}$, O G $= \rho \cos a$, C O $= \sqrt{r^2 - \rho^2 \sin^2 a} + \rho \cos a$, A O $= r + \rho$, and $\therefore$ the value of A C, or, A O — C O, is

$$\begin{aligned} x &= r + \rho - (\sqrt{r^2 - \rho^2 \sin^2 a} + \rho \cos a) \\ &= \rho\,(1 + \cos a) + (r - \sqrt{r^2 - \rho^2 \sin^2 a}) \;\ldots\; (1) \end{aligned}$$

By differentiating this expression of the distance traversed by C, and dividing by $d\,t$, we obtain the velocity of the extremity C, viz.,

$$\frac{d\,x}{d\,t} = \frac{\rho\,d\,a}{d\,t} \sin a \left(\frac{\rho \cos a}{\sqrt{r^2 - \rho \sin^2 a}} + 1 \right);$$ that is,

$$v = v' \sin \alpha \left(\frac{\rho \cos \alpha}{\sqrt{r^2 + \rho \sin^2 \alpha}} + 1 \right) \ldots\ldots\ldots (2)$$

But, if P denote the moving force acting at C in the direction C B, P′ the effective force of D, and c the angle D C O; then the force P, transferred in the direction of the connecting rod, becomes P sec c, and this resolved in the direction of the tangent to the circle $= \mathrm{P} \sec c\, (c + \alpha) =$

$\mathrm{P}\, (\tan c \cos \alpha + \sin \alpha)$; also $\tan c = \dfrac{\rho \sin \alpha}{\sqrt{r^2 - \rho^2 \sin^2 \alpha}}$;

$$\therefore \mathrm{H}' = \mathrm{P} \sin \alpha \left(\frac{\rho \cos \alpha}{\sqrt{r^2 - \rho^2 \cos^2 \alpha}} + 1 \right) \ldots\ldots (3)$$

Equation (2) hence reduces to $v = v' \dfrac{\mathrm{P}'}{\mathrm{P}}$, and, therefore,

$$\mathrm{P}\, v = \mathrm{P}'\, v' \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

that is, the moving *power* at C is always equal to the effective power at D.

The moving force P being nearly uniform, the force P′ expressed by the equation (3) will be materially different at different positions of the crank, and the velocity of the engine will, in consequence, be subject to small fluctuations in the course of each revolution. To ascertain the law of these variations, it will be necessary to have recourse to the usual equation of rotatory motion. Let R denote the real moment of the resistance, including friction, reduced to the point D, or the force, which uniformly applied along the circumference E D would just suffice to preserve the mean velocity of the engine without any variation; then P′ being the force actually applied, the effective accelerating force, or the part tending to produce acceleration, will be P′ — R; and where this is negative, the velocity $\dfrac{d\,\alpha}{d\,t}$ must be retarded instead of accelerated. If we now multiply the force P′ — R by ρ, the leverage at which it acts, the product $\rho\,(\mathrm{P}' - \mathrm{R})$ is its moment or tendency to generate angular motion. Hence, if M designate the moment of the inertia to be overcome, and if we neglect, as comparatively insignificant, the slight varia-

tions of the resistance R due to the small changes of velocity; and suppose it to continue uniform, we shall have

$$M \frac{d^2 a}{d t^2} = \rho (P' - R);$$

this being multiplied by $\frac{2\, d a}{d t}$, there results

$$M \frac{2\, d a\, d^2 a}{d^2 t^2} = 2 P' \rho \frac{d a}{d t} - 2 R \rho \frac{d a}{d t}$$

$$= 2\left(P' v' - R \rho \frac{d a}{d t}\right)$$

$$= 2\left(P v - R \rho \frac{d a}{d t}\right)$$

$$= 2\left(P \frac{d a}{d t} - R \rho \frac{d a}{d t}\right).$$

Consequently, by integration,

$$M \left(\frac{d a}{d t}\right)^2 = 2 (P c + P x - R \rho a),$$

$$\therefore \frac{d a}{d t} = \sqrt{\frac{2}{M} (P c + P x - R \rho a)} \ldots\ldots\ldots (5)$$

Now the engine being supposed to have acquired her permanent speed, there can be no progressive acceleration, the same velocity must recur at the period of each revolution. The velocity at E is obtained by putting $x = 0$, $a = 0$, and is

$$\left(\frac{d a}{d t}\right)' = \sqrt{\frac{2}{M} P c} \ \ldots\ldots\ldots\ldots (a)$$

Also, the velocity at H is obtained by putting $x = 2 \rho$, $a = \pi$, and is

$$\left(\frac{d a}{d t}\right)'' = \sqrt{\frac{2}{M} (P c + 2 P \rho - R \rho \pi)} \ldots\ldots (b)$$

Again, it is to be observed that the preceding investigation equally applies to returning motion along H D°, pro-

vided, in that case, the symbol α represents the angle HOD°, and x', which is accentuated for distinction, the retrograde distance BC. If $\therefore$ $\alpha\, x'$, being each $= 0$, the velocity at H will be

$$\left(\frac{d\,\alpha}{d\,t}\right)'' = \sqrt{\frac{2}{\mathrm{M}}\,\mathrm{P}\,c'} \;\ldots\ldots\ldots\ldots\; (c)$$

and when $\alpha = \pi$, $x' = 2\,\rho$, the recurring velocity at E will be

$$\left(\frac{d\,\alpha}{d\,t}\right)''' = \sqrt{\frac{2}{\mathrm{M}}\,(\mathrm{P}\,c' + 2\,\mathrm{P}\,\rho - \mathrm{R}\,\rho\,\pi)} \ldots\ldots\; (d)$$

Thus it appears that the squares of the three velocities $\left(\frac{d\,\alpha}{d\,t}\right)'$, $\left(\frac{d\,\alpha}{d\,t}\right)''$, $\left(\frac{d\,\alpha}{d\,t}\right)'''$ are in arithmetical progression, and that this progression would go on indefinitely if the resistance R did not augment so as to destroy the common difference $2\,\mathrm{P}\,\rho - \mathrm{R}\,\rho\,\pi$, and cause the same velocity to recur at each period. It hence follows that, so long as the smallness of R renders $2\,\mathrm{P}\,\rho - \mathrm{R}\,\rho\,\pi$, or $2\,\mathrm{P} - \mathrm{R}\,\pi$, positive, the engine will be acquiring additional speed; and when R becomes such that $2\,\mathrm{P} - \mathrm{R}\,\pi = 0$, the power will just be capable of maintaining unaltered the periodical movement, and the general speed will not admit of any further increase. For permanent speed we must therefore have

$$\mathrm{R} = \frac{2}{\pi}\,\mathrm{P} \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\; (6)$$

This result shows that the general effect of the moving force P, acting obliquely on OD at D, is the same as if $\frac{2}{\pi}$ P, or very nearly $\frac{7}{11}$ P, were always applied perpendicularly at the same point.

It is not to be inferred from this that the effect of any portion of the power is lost. The quantity of power developed by a constant force is measured by the force multiplied into the distance through which it has acted. In a semi-revolution, the power developed by P is hence $\mathrm{P} \times 2\,\rho$; that expended by R would similarly be $\mathrm{R} \times \pi\,\rho$; and,

according to the equation (6), it appears that the expenditure would be the same in both cases, and that no loss is sustained through the obliquity of the action of the crank, except that which arises from the additional friction caused by the stress on the shaft.

Substitute the value of R from (6) in (5), and there results, for permanent speed, the angular velocity of the shaft of the engine at any period of the stroke, the following formula,

$$\frac{d\,a}{d\,t} = \sqrt{\frac{2\,\mathrm{P}}{\mathrm{M}}\left(c + x - \frac{2\,\rho}{\pi}\,a\right)} \quad \ldots\ldots\ldots \ (7)$$

and for any position D from E to H, if β denote the angle E O D, this becomes

$$\omega = \sqrt{\frac{2\,\mathrm{P}}{\mathrm{M}}\left\{c + \rho(1 - \cos\beta) - \frac{2\rho}{\pi}\beta + \left(r - \sqrt{r^2 - \rho^2\sin^2\beta}\right)\right\}}\,. \ (8)$$

For a point D° in the returning half stroke from H to E, if β denote the angle H O D°, the value of A C′ is obtained by substituting $\pi - \beta$ for a in the value of x expressed by equation (1), and is therefore

$$\mathrm{A\,C} = \rho\,(1 + \cos\beta) + \left(r - \sqrt{r^2 + \rho^2\sin^2\beta}\right);$$

and if we deduct this from the whole distance $\mathrm{A\,B} = 2\,\rho$, we obtain the expression for B C or x', viz.,

$$x' = \rho\,(1 - \cos\beta) - \left(r - \sqrt{r^2 - \rho^2\sin^2\beta}\right) \quad \ldots\ldots \ (9)$$

Therefore by equation (7) the velocity at D° is

$$\omega^{\circ} = \sqrt{\frac{2\mathrm{P}}{\mathrm{M}}\left\{c + \rho(1 - \cos\beta) - \frac{2\rho}{\pi}\beta - \left(r - \sqrt{r^2 - \rho^2\sin^2\beta}\right)\right\}}\,. \ (10)$$

The velocities ω' ω'' at D′ D″ will now be determined by substituting $\pi - \beta$ for β in equations (8) and (10); thus we find

$$\omega' = \sqrt{\frac{2\mathrm{P}}{\mathrm{M}}\left\{c - \rho(1 - \cos\beta) + \frac{2\rho}{\pi}\beta - \left(r - \sqrt{r^2 - \rho^2\sin^2\beta}\right)\right\}}\,. \ (11)$$

$$\omega'' = \sqrt{\frac{2\mathrm{P}}{\mathrm{M}}\left\{c - \rho(1 - \cos\beta) + \frac{2\rho}{\pi}\beta + \left(r - \sqrt{r^2 - \rho^2\sin^2\beta}\right)\right\}}\,. \ (12)$$

To put these in a more convenient form for comparison, let (ω) denote the velocity at E, as expressed by equation (a), and assume

$$\left.\begin{aligned} A &= \rho\left(\frac{2}{\pi}\beta - 1 + \cos\beta\right) \\ h &= r - \sqrt{r^2 - \rho^2 \sin^2\beta} \end{aligned}\right\} \text{.........} \quad (13)$$

Then the angular velocities at any four corresponding points D°, D, D′, D″, will be

$$\left.\begin{aligned} \omega^{\circ} &= \sqrt{(\omega)^2 - \frac{2\,P}{M}(A + h)} \\ \omega &= \sqrt{(\omega)^2 - \frac{2\,P}{M}(A - h)} \\ \omega' &= \sqrt{(\omega^2) + \frac{2\,P}{M}(A - h)} \\ \omega'' &= \sqrt{(\omega)^2 + \frac{2\,P}{M}(A + h)} \end{aligned}\right\} \text{.........} \quad (14)$$

which are here arranged respectively in the order of their magnitudes, that at D° being the least, and that at D″ the greatest.

These formulæ may be put in a still more convenient form by neglecting the higher powers of the very small variations; we shall then have

$$\omega^{\circ} = (\omega) - \frac{P}{M\,(\omega)}(A + h),$$

$$\omega = (\omega) - \frac{P}{M\,(\omega)}(A - h),$$

$$\omega' = (\omega) + \frac{P}{M\,(\omega)}(A - h),$$

$$\omega'' = (\omega) + \frac{P}{M\,(\omega)}(A + h).$$

But for h we may now substitute $\frac{\rho^2}{2r}\sin^2\beta$, and there will result

$$\left.\begin{aligned}\omega^\circ &= (\omega) - \frac{P}{M(\omega)}\left(A + \frac{\rho^2}{2r}\sin^2\beta\right)\\ \omega &= (\omega) - \frac{P}{M(\omega)}\left(A - \frac{\rho^2}{2r}\sin^2\beta\right)\\ \omega' &= (\omega) + \frac{P}{M(\omega)}\left(A - \frac{\rho^2}{2r}\sin^2\beta\right)\\ \omega'' &= (\omega) + \frac{P}{M(\omega)}\left(A + \frac{\rho^2}{2r}\sin^2\beta\right)\end{aligned}\right\} \quad \ldots\ldots (15)$$

in which the values of A in parts of ρ are from (13) found to be as follows,

β	A	β	A	β	A
0°	0·0000	40°	+ ·2080	70°	+ ·1198
10	+ ·0959	45	+ ·2071	75	+ ·0922
20	+ ·1619	50	+ ·1984	80	+ ·0625
30	+ ·1994	60	+ ·1667	90	+ 0·0000

From these last results we conclude that the velocity (ω), which occurs at E and H, is the *mean velocity*, and that the greatest and least velocities occur in the quadrants of D″ and D°, at the points where

$$A + \frac{\rho^2}{2r}\sin^2\beta, \text{ or } \frac{2}{\pi}\beta + \cos\beta + \frac{\rho^2}{2r}\sin^2\beta,$$

attains its maximum value. If we assume these points to be the middle of the respective quadrants, so that $\beta = 45°$,—which will be very nearly the case with all the proportions observed in practice, and cannot sensibly affect the accuracy of the results,—the greatest and least velocities will be $(\omega) \pm \frac{P\rho}{M(\omega)}\left(\frac{\rho}{2r} + 0{\cdot}207\right)$, and the deviation from the mean value will be

$$\frac{P\rho}{M(\omega)}\left(\frac{\rho}{2r} + 0{\cdot}207\right).$$

Again, by substituting the values of P′, R, the force P′ − R, which at each position tends to accelerate motion, is found to be

$$\mathrm{P}' - \mathrm{R} = \mathrm{P}\left\{\sin a\left(\frac{\rho \cos a}{\sqrt{r^2 - \rho^2 \sin^2 a}} + 1\right) - \frac{2}{\pi}\right\}. \quad (16)$$

It hence appears that the motion is

$$\begin{matrix}\text{accelerated}\\ \text{retarded}\end{matrix}\left\{\text{when } \sin a\left(\frac{\rho \cos a}{\sqrt{r^2 - \rho^2 \sin a}} + 1\right) \text{is} \left\{\begin{matrix}\text{greater}\\ \text{lesser}\end{matrix}\right\} \text{than } \frac{2}{\pi}\right.$$

In the preceding investigation we have only considered the action of a single crank; but it will be found to apply, with a slight modification, when the rotatory motion of the shaft is maintained by the action of any number of cranks making given angles with each other. We have only to substitute, in place of P′, in the right-hand member of the equation of motion, the sum of the forces P′, arising from the several cranks; and, by following out the same process, formula (5) will become

$$\frac{d a}{d t} = \sqrt{\frac{2}{\mathrm{M}}\left\{\mathrm{P}\, c + \Sigma\, (\mathrm{P}\, x) - \mathrm{R}\, \rho\, a\right\}},$$

in which $\Sigma\,(\mathrm{P}\,x)$ includes the same term for each crank. Let (ω) denote the value of this angular velocity, when $x = 0$, $a = 0$, and the formula, for any other position, will become

$$\omega = \sqrt{(\omega)^2 + \frac{2}{\mathrm{M}}\left\{\Sigma\,(\mathrm{P}\, x) - \mathrm{R}\, \rho\, a\right\}} \ldots\ldots (17)$$

By attending to the nature of the integration, it is evident that this formula applies generally to any assigned position throughout the entire period of each revolution of the shaft, if we give to the symbols x, a the following signification, *i. e.*, $x =$ the entire distance travelled over by the extremity c, for each crank; and $a =$ the angle described by the revolution of each shaft, each being estimated from that position in which the velocity was (ω).

Let us take the practical case, in which the revolution of

the shaft is maintained by two or more cylinders, and in which the equal arms of the two or more cranks are placed at right angles to each other, so that each of them may be in full action when the other is "on its centre." If we suppose one entire revolution to be performed, and that there are two cylinders, the distance x, traversed by the extremity of each connecting rod, will be $4\,\rho$, the angle a will be $2\,\pi$, and the velocity will become $\omega =$

$$\sqrt{(\omega)^2 + \frac{2}{M}(4\,P\,\rho + 4\,P\,\rho - 2\,R\,\rho\,\pi)} =$$

$$\sqrt{(\omega)^2 + \frac{4\rho}{M}(4\,P - R\,\pi)}.$$

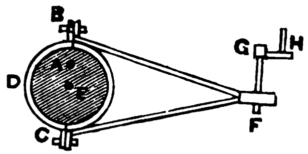
When the engine has attained its regular speed, this velocity must correspond with (ω , since precisely the same motions must recur in successive revolutions; we must hence have $4\,P = R\,\pi = 0, \therefore R = \frac{4}{\pi}P$; which value is the same as $\frac{2}{\pi}(2\,P)$, in which $2\,P$ is now the moving power, and we find it corresponds with formula (6) for the single crank.

We have gone into an investigation of the motion of the crank, as it forms one of the most important appendages of the steam engine, and, previous to the publication of the second edition of Tredgold's work on the steam engine, had met with very little attention from scientific writers. It is doubtless the most simple, and perhaps the most efficient, contrivance that can be devised to convert a reciprocating action into a rotatory motion; and in this respect we cannot be surprised that it has not been superseded by any one of the numerous inventions which have been proposed with the view of dispensing with it.

THE ECCENTRIC.

The eccentric is a circular disc revolving within a strap or ring, and having its axis of revolution on one side of its centre. It is used for giving a reciprocating motion to the slide-valve or to the feed-pump of a steam engine.

In the annexed Fig. E is the centre and A the axis the eccentric wheel, which is always fixed on the axis of tl fly wheel of the stationary engine; a hoop B C D embrace this wheel so as just to allow i to turn freely within its circle the hoop being generally oi two pieces joined at B and C; a frame B F C connects the hoop with the extremity F of the bent lever F G H, which turns on the centre G. When E revolves on its eccentric axis A, the frame B F C will be drawn alternately to the right and left, and the end, F, of the bent lever F G H will describe at every revolution two arcs of a circle; the reciprocating motion of F, thus produced, transmits a like kind of motion to the other end of the lever, to which the slide-valve of the engine is attached.



"*The eccentric arm or crank is by far the most simple mode of converting rotation into reciprocation*, and it has the valuable property of beginning the motion in each direction gently, and again gradually retarding it so as to avoid jerks. Nevertheless, the law of variation in the velocities is not always the best adapted to the requirements of mechanism; but the reciprocation is produced so simply, that it is often worth while to retain the crank, and correct the law of velocity by combining other pieces with it in a train. By trains of link-work very complex laws of motion may be derived from a uniformly revolving driver."—*Professor Willis.*

To Construct an Eccentric Wheel.

From the centre of the shaft A take A E equal half the length f the stroke which the wheel is intended to work; and from E s a centre, with a radius somewhat greater than E D, describe circle, which will represent the required wheel.

Example.—Suppose an eccentric wheel is required to ve a stroke of 16 inches, the diameter of the shaft being nches, and the thickness of metal for keying it on to the ft 2 inches; then set off, from A to E, $16 \div 2 = 8$ inches,

$8 + \frac{6}{2} + 2 = 13$ inches, the radius of the eccentric

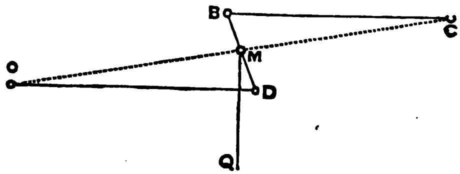
el required.

Let S = space the end F is moved through by the eccentric wheel, and s = space through which the slide moves; then $FG \times s = GH \times S$. If in this equation any three of the parts be given the fourth may be found.

PARALLEL MOTION.

This simple and beautiful arrangement of link-work was invented by the celebrated Watt, to convert the reciprocating circular motion of the extremity of the great beam of the steam engine into a reciprocating rectilinear motion adapted to the piston rod.

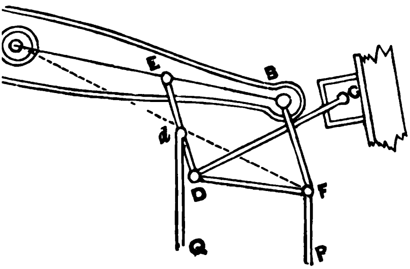
Let the two equal rods C B, O D, which are connected by a third rod or link B D, move on their fixed centres C O;



and let M be the middle of B D. Now, if C B be made to move alternately upwards and downwards, which will cause O D to move on its centre O in the same manner, it will be found that the point M will ascend and descend in a line which will not deviate sensibly from a vertical straight line. For when the point B is moved upwards, the upper extremity of the rod B C is drawn a little to the *right*, at the same time extremity D of the rod O D is drawn a little to the *left*. When the extremity B descends, the extremity D also descends; thus the two extremities are again drawn, the one a little to the *right*, and the other a little to the *left*. It will be readily understood that while the ends of the rod or link B D are thus alternately made to move right and left, its middle point M will not sensibly deviate to the right nor to the left, but will move upwards and downwards in a line not sensibly varying from a vertical direction. When BC=DO, the point M falls in the line C O.

The complete parallel motion, which is most universally adopted in large steam engines, is shown in the annexed Fig. When so employed the beam of the engine becomes

one of the radius rods of the system. A B is half this beam, of which the centre of motion is A. It has two links E D, B F, jointed to it, of which B F is termed the *main link*, and E D the *back link;* and these are connected below by a third link D F, called the *parallel rod*, and equal to B E. The radius rod or *bridle rod* C D is jointed to the end D of the back link E D, and its centre C is fixed at a vertical distance below A equal to E D or B F. The length of the rods are so proportioned that F shall be the point to which the rectilineal motion is communicated, or *parallel point*, as it is termed. See Professor Willis's "Mechanism," Art. 447, who deduces from a learned and abstruse investigation the following simple equation, exhibiting the proportions of the parts constituting the parallel motion,



$$C D \times D F = A E^2$$

$$\text{or} \quad C D = \frac{A E^2}{D F} = \text{length of the radius rod.}$$

Since the parts A E, E D, C D, considered separately, form a system similar to the arrangement in the preceding Fig., it follows that if the proper point d between D and E be taken, an additional parallel motion will be obtained; so that this form combines two parallel motions in one, and is commonly so employed in steam engines, by suspending the great piston rod P F from F, and the air-pump rod Q d from d in the link E D. The position of the point d is found from the following formula,

$$D d = \frac{E D \, . \, A E}{A E + E D} \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

If the position of the point d be given, the length of the

radius rod C D may be found by transposing the preceding formula, which gives

$$C D = \frac{A E (E D + D d)}{D d} \text{ .... } (2)$$

When the lengths of the several parts of the parallel motion, as E D, A E, C D, are given, the distance D d may be readily found by formula (1); and when A E, E D, D d are given, the length of the radius rod C D may be found by formula (2).

Note.—If the system of link-work in the parallel motion be moved into all the positions it is capable of taking, the actual paths of the points F and d would be found to be (since the extent of the stroke of the piston is small) in curves in the shape of the figure 8; but the small portions of the curves actually described differ insensibly from right lines. See Professor Willis's "Mechanism," Arts. 441 to 452, where various other important investigations on the same subject are given. Professor Hann, in his works on the steam engine, and in his edition of "Tredgold," has also given various important methods of constructing the parallel motion adapted to particular cases; but the methods here given are those most extensively used.

PADDLE WHEELS.

The floats of the paddle wheel may be considered as a series of levers coming into successive action upon the water. Each arm of the wheel to which a float is attached represents one lever; the reaction of the water on the paddle board or float constitutes the fulcrum; the resistance is that of the water to the motion of the vessel at the centre of the wheel; and the power is that produced by the engine. As the fulcrum is produced by the reaction of the water on the paddle board (as there can be no reaction if the surface of the paddle board does not move through the water), the true fulcrum must be at that point on which, if simply immersed, there would be no reaction. This point, moving in the direction of the surface of the paddle board, can meet with no resistance from the water; some power must therefore be exerted to force the paddle boards through the water, in addition to that which would be required to propel the vessel. Hence a *statical theory* may be deduced for the power requisite to move the floats of the paddle wheel.

ON THE DIFFERENT KINDS OF PADDLE WHEELS.

1. *Field's Paddle Wheel,* which is called the cycloidal wheel, differs from the common paddle wheel in the arrangement of the paddle boards, and was exhibited before the Lords of the Admiralty in 1833. However, Mr. Galloway took out a patent for this same invention in 1835; but the credit of it clearly belonged to the first-named gentleman.

Mr. Field thus describes his wheel in the *London Journal* for December, 1835:—

"Each board is divided into several parts, or narrower boards, and arranged in, or nearly in, such cycloidal curves that they all enter the water in immediate succession, thus avoiding the shock produced by the common board, so unpleasant to passengers, injurious to the vessel, and wasteful of the power. As the acting face of each board is radiating, it propels while passing under the centre in the ordinary way, and when it emerges the water escapes simultaneously from each narrow board, and, consequently, cannot lop-up."

Mr. Barlow gives the following description of these wheels:—

"The principle of this contrivance consists in dividing the paddle into a number of parts, which are placed upon the wheel in the curve of a cycloid, so that they enter the water at the same spot and follow one another so rapidly as to cause little resistance to the engine on entering the water, and afterwards separate so as to afford full scope for their action in passing the centre, and in coming out allow the water to escape readily from them."

There is some difference of opinion respecting the advantages and disadvantages of these wheels. Mr. Barlow says that "the use of the cycloidal wheel is very likely to become general, and supersede that of Morgan, from its superior strength and simplicity, while it does away with most of the evils to which the common wheel is subject."

Mr. Mornay, on the contrary, seems rather to underrate it. His opinion is, "that it remains to be decided by experiment, whether the disadvantages of this wheel are overbalanced by the advantages it *seems* to possess."

This wheel has been improved since the time of its first introduction, in the following rather remarkable manner. At the time of the adoption of this wheel, it was with six or *seven bars in each set,* instead of the common paddle boards

This number has been much reduced (indeed the wheels now used in her Majesty's service have only two boards in a set), as every reduction was found to be more advantageous. However, these improvements cannot be carried on farther as in that case they would return to the common wheel again.

The *Great Western* and *British Queen* are among the vessels that have been fitted with these wheels.

2. *Paddle Wheels with Oblique Floats.*—One of the most simple of this kind is that of Mr. S. Hall, for which he obtained a patent in 1836. Here the paddle boards are placed obliquely to the rims and the axis of the wheel. The subject of the patent, however, is not the use of the oblique floats, but the making one half of them to enter the water in one diagonal direction, and the other in the reverse.

The objections are (1), that it requires a greater surface of paddle board to produce the same effect, for Mr. Mornay proves that the power in these wheels must exceed that in common wheels in the ratio of $\frac{1}{10}$; and (2), that the shock is very little reduced, and the loss of power from oblique action is greater than with the common wheel.

3. *Morgan's Paddle Wheel with Feathering Floats.*—The construction of this wheel is as follows. The paddles turn on spindles, having a bearing on the framework and on the wheel, which is polygonal, having as many sides as there are paddles. The inside frame or polygon alone is attached to the shaft of the engine, which does not continue beyond the side of the vessel; and the outer one has an independent bearing on a centre attached to the paddle box, so that it derives its motion entirely from the arms or angles of the polygon, the space between the two frames being left entirely free. A crank is fixed to the paddle box, on which the outer polygon revolves; it projects in an inclined direction in the open space between the sides of the wheel. Each paddle has a crank attached to it at an angle of about 70°, and arms connect the extremities of the cranks with a moveable boss, which revolves on a fixed centre; one of these arms is fixed to the boss, and is called the dividing arm.

This wheel is quite free from back-water, and from any shock, as both the upper and lower edges of the float as

nearly tangents to their respective cycloids at the time of entering the water. These wheels have always been praised for their beautiful action, their strength, and their durability, as well as for the safety, comfort, and economy with which they are attended. They have been adopted in many government and mercantile steamers, and always with success. It has been found by experiments, that in light immersions little advantage is gained by these wheels, since, reckoning the power of the engine 1, the effective power = ·660 in the common wheel, and ·666 in Morgan's. This arises from the loss from the additional velocity required to obtain the resistance or receding of the vessel, being fully equal to that of oblique action in the common wheel. In cases of deep immersion the result is different; the effective power of the common wheel will, in a very deep immersion, equal only ·553, while that of Morgan's wheel remains the same.

In the *Firebrand* steamer, Morgan's wheel, with less power and less float, gave a greater speed than the common wheel. We may, therefore, conclude that this wheel has the superiority in sea, and the common wheel in river navigation.

There are other modifications of the paddle wheel with feathering floats, by Buchanan and Oldham; but they have been found to possess no advantages at all comparable to Morgan's, just described, and have, therefore, never been adopted in practice.

To find the Centre of Pressure in Morgan's Paddle Board, &c.

Put R and r = the radii of the wheel and rolling circle respectively, and d = the depth of the paddle board; then the depth of the centre of pressure from the upper edge of the paddle board, as deduced from Mr. Barlow's formula, is

$$\sqrt[3]{\frac{((R - r) + d)^4}{4d}} - (R - r).$$

Example.—The radius of the wheel of the *Messenger* steam vessel is $9\frac{2}{3}$ feet, the radius of the rolling circle is $6\frac{2}{3}$ *feet, and* the depth of the paddle board 2 feet; required *the diameter of the* centre of pressure.

By the formula,

$$\sqrt[3]{\frac{(9\frac{2}{3} - 6\frac{2}{3} + 2)^4}{4 \times 2}} - (9\frac{2}{3} - 6\frac{2}{3}) = \sqrt[3]{\frac{625}{8}} - 3 =$$

$\frac{5}{2}\sqrt[3]{5} - 3 = \frac{5}{2} \times 1{\cdot}70976 - 3 = 1{\cdot}2744$ feet = distance of the centre of pressure from the upper edge of the paddle. Now, as the radius of the wheel is $9\frac{2}{3}$ feet, and the depth of the paddle 2 feet, the distance from the centre of the wheel to the upper edge of the paddle is $9\frac{2}{3} - 2 = 7\frac{2}{3}$ feet; then $7\frac{2}{3} + 1{\cdot}2744 = 8{\cdot}941$ = distance of the centre of the wheel to the centre of pressure, and $8{\cdot}941 \times 2 = 17{\cdot}882$ = diameter of the centre of pressure.

The speed of the vessel was found to be $9\frac{3}{4}$ miles per hour; therefore $\frac{9\frac{3}{4} \times 5280}{60} = 858$ feet per minute $= 14{\cdot}3$ per second.

Since the diameter of the centre of pressure, or effective diameter of the wheel, is now known, the velocity of the wheel in excess of that of the vessel, or that at which it recedes in the water to produce the resistance necessary for propelling the vessel, may be found by the following method:—

Multiply the velocity by the area of the paddle board, and by $62\frac{1}{2}$ lbs. (the weight of a cubic foot of water), and divide by twice the force of gravity at the earth's surface, which is $64\frac{1}{3}$; this quotient, multiplied by the velocity of the wheel, will give the power expended on the paddles; and this, being divided by the whole power of the engine, will give the proportion consumed on the vertical paddle.

The diameter of the centre of pressure being $17{\cdot}882$ feet, its circumference is $17{\cdot}882 \times 3{\cdot}1416 = 56{\cdot}2$, and this, multiplied by 22, the number of strokes made by the engine per minute, gives $56{\cdot}2 \times 22 = 1236{\cdot}4$ = velocity of the centre of pressure per minute, or $1236{\cdot}4 \div 60 = 20{\cdot}60$ = velocity per second.

The velocity of the rolling circle, as above, is $14{\cdot}3$ feet per second; hence $20{\cdot}60 - 14{\cdot}3 = 6{\cdot}30$ feet per second, which is the excess of the velocity of the wheel over that of the vessel; and $(6.30)^2 \times \frac{62\frac{1}{2}}{64\frac{1}{3}} \times$ area of the paddle board = *pressure in pounds* on the vertical paddle; and, since the pad

board is 10 feet long and 2 broad, its area is $10 \times 2 = 20$ square feet; hence,

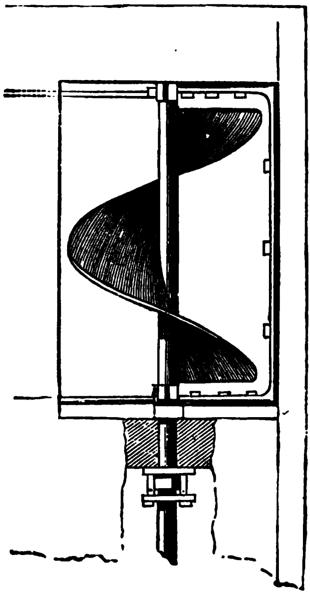
$$\frac{(6{\cdot}30)^2 \times 62\frac{1}{2} \times 20}{64\frac{1}{3}} = 771 \text{ lbs., nearly,}$$

The velocity of the centre of pressure, found above, is 1238·6 feet per minute;

$$\therefore \frac{2 \times 1238{\cdot}6 \times 771}{33{\cdot}000} = 57{\cdot}77 \text{ horse-powers, nearly.}$$

MARINE SCREW PROPELLERS.

Screw propellers for navigation, by means of steam power, have now become objects of great importance to all nations;

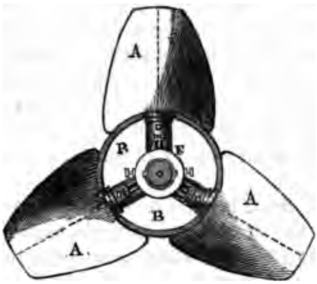


they are especially applicable to vessels of war, the machinery *for propulsion being* placed so deep in the hold of the ship

as to secure it completely from the reach of gunshot. Screw propellers, however variously they may be modified, all derive their power of propelling by being placed on an axis which is parallel to the keel, and by having threads or blades extending from the axis, which form segments of a helix, or spiral; so that by causing the axis to revolve, the threads or blades worm their way through the water, much in the same way as a carpenter's screw inserts itself into a piece of wood, with the difference in the case of the screw-propeller of its making the water recede. Screw propelling is not of recent invention: M. Duquet in 1727, and Mr. Pancton in 1768, both produced machinery of this kind; other inventions followed until a recent date.

When the *Archimedes* was first tried down the river Thames in 1836, the shape of the screw was as represented in the figure, *i. e.*, a single thread of thin sheet iron, bent to fit sixteen wrought-iron arms fixed round the axis at equal distances, so as to form a helix, or screw. This screw, from its having only one thread of long pitch, viz., 45°, caused a great commotion in the water, and a great deal of vibration in the stern of the vessel, so that on reaching Sheerness the vessel was laid ashore, and portions of the iron plate taken off at equal intervals; but the effect was not improved. The form of the screw was then changed to a double one; and, finally, a series of experiments was made by order of the Admiralty, with three or four-bladed screws, which ended in the adoption of the *two*-bladed screw. These experiments were tried in H. M. ships *Dwarf* and *Rattler*, till a speed had been produced in the *Dwarf* of 12¼ knots per hour with Mr. Rennie's conoidal screw of cast-iron, being the greatest speed which had ever been attained by the screw up to that time. The speed attained by the *Archimedes* was about nine knots per hour, which, taken as a first experiment, was a great performance. The *Archimedes* beat most of the fastest steamers then known, and made a voyage all round Great Britain, being a distance of 2096 nautical miles, in 237½ hours, or nearly nine knots per hour through the whole distance. The *Archimedes* afterwards sailed from Plymouth to Oporto in 69 hours, and returned from thence, against strong head-winds and high seas, in 88 hours; and the reports of some of the most eminent officers predicted all that has since been realised by the screw.

Griffith's Screw Propeller.—This is probably one of the best modifications of the marine screw which has yet been produced. Its chief peculiarity consists in a large hollow sphere B being substituted for the central portion of the blades of the ordinary screw, the blades A, A, A terminating upon the sphere instead of upon the usual boss. It has been found by experiment that the central portion of the blades of the common screw absorbs nearly 20 per cent. of the propelling power, without giving out any useful effect, in consequence of that portion of the blades being nearly in a line with the shaft, so that the water is thrown off at right angles, thus disturbing the more solid water upon which the outer and more effective portion of the screw is about to act. The use of the sphere is, therefore, to fill up the space otherwise occupied by the central portion of the screw blades, a smooth globe of this kind revolving in the water with comparatively little friction or resistance.



It is found also that the vibrations caused by the revolutions of the common screw are considerably lessened by this contrivance. Another feature in Griffith's propeller consists in the peculiar form of the blades, which, unlike the common screw, are larger towards the sphere and smaller towards the extremities: the diameter of the sphere being one-third of the whole diameter of the screw, the breadth of each blade at the root is equal to the full diameter of the sphere, tapering to two-thirds of this breadth at the extremities of the blades. The object in reducing the blades at their ends is to compensate for the varying velocity with which the different parts of their length travel through the water.

There is still another feature in this propeller, *i. e.*, the power of altering the pitch or angle of the blades by a suitable arrangement of apparatus contained within the hollow sphere; or, if it be wished to stop the engines, and sail *under canvas, the blades* may be feathered (as in Maudslay's

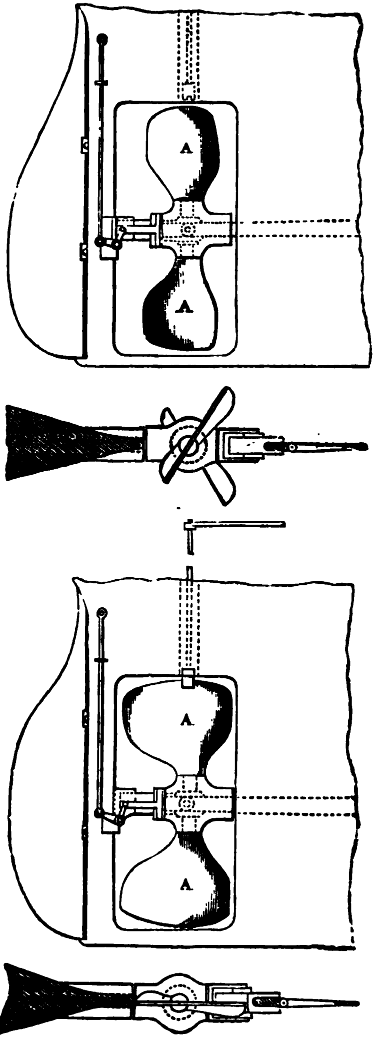
feathering screw), so as to present their edges to the water, and thus cause the least amount of resistance to the ship.

"*Maudslay's Feathering Screw.*—This screw is represented on the following page. The object sought to be obtained is, that the blades, whenever the vessel is put under canvas, and the screw not required, should be placed in a direction parallel with the line of the keel, and so form, as it were, a portion of the dead wood, as they cause considerable obstruction if they be allowed to remain fixed in their position, or even though they be disconnected from the engine and allowed to revolve. In auxiliary sailing vessels, not fitted with a trunk or aperture for the purpose of raising the screw out of the water, this is particularly valuable; but it will also be found useful in ships of war, by lessening the width of the trunk through which it has to rise, if this be desired, and also by the facility which it gives in emergencies, for placing the vessel under canvas or under steam, as well as for placing the vessel under canvas when it may not be possible to keep the engines at work for want of fuel or other causes. The blades are locked in the vertical position by means of a catch and a screw, passing through the stern of the vessel, as shown in the figure."

"The *Diameter of the Screw* should, in most cases, be made as great as the draught of water will admit, and for sailing in smooth water its upper edge need not be more than a few inches below the surface. In the case of sea-going vessels, it is preferable to keep it 1½ or 2 feet below the mean surface of the water."

"*The Area of the Screw.*—By the area of the disc of the screw is understood the area of the circle described by the extremity of its diameter. When the area of the blades is spoken of, their actual oblique surface should always be especially distinguished from the plane projection of the resisting surface. This latter measurement, as representing the actual amount of surface directly employed in the propulsion of the vessel, is probably the most important of these areas."—*Murray on the Marine Engine, Weale's Series.*

1st. In position for use as a propeller.
2nd. In position for sailing under canvas alone.



THE THEORY OF THE SCREW.

Various authors have given their theoretical deductions for determining the power of the screw. The following is from Hann and Gener's "Treatise on the Steam Engine."

The surface of the screw is generated by a line normal to the axis, round which it revolves uniformly, to the extent of one revolution.

Let v = velocity of the vessel in feet per second,
w = angular velocity of the screw,
h = length or pitch of the screw,
r = radius of any point P,
a = angle of inclination of the surface of the screw at that point, with a plane section perpendicular to the axis; and let

$$k = \frac{h}{2\pi} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

$$u = k\,\omega = \frac{h}{2\pi}\,\omega \quad \ldots\ldots\ldots\ldots \quad (2)$$

Then, $$\tan a = \frac{h}{2\,r\,\pi} = \frac{k}{r} \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

Effective angular velocity

$$= \frac{r\,\omega \tan a - v}{\tan a} = \frac{r}{k}\,(k\,\omega - v) = \frac{r}{k}\,(u - v).$$

Velocity perpendicular to the screw

$$= \left(r\,\omega - v\,\frac{\cos a}{\sin a}\right)\sin a$$

$$= (r\,\omega \tan a - v)\cos a = (u - v)\cos a.$$

Elementary surface at P $= \dfrac{2\,\pi\,r}{\cos a}\,.\,d\,r;$

∴ the elementary pressure

$$d p = \frac{W}{2\,g}\cdot\frac{2\,\pi\,r}{\cos a}\cdot d\,r\,(u - v)^2 \cos^2 a$$

$$= \frac{W}{g}\cdot \pi\,r\,d\,r \cos a\,(u - v)^2.$$

Let $$C = \frac{W}{g} \cdot 2\pi (u - v)^2 \;\ldots\ldots\; (4)$$

and $dp = C r\, dr \cos a = \dfrac{C r^2\, dr}{\sqrt{k^2 + r^2}}$, from formula (3);

$$\therefore\ d\,(\text{effective power}) = v\,(dp \cos a) = C v \frac{r^3\, dr}{k^2 + r^2}.$$

But $$\int \frac{r^3\, dr}{k^2 + r^2} = \int \left(r\, dr - \frac{k^2 r\, dr}{k^2 + r^2} \right)$$
$$= \tfrac{1}{2} \left(r^2 - k^2 \log \frac{k^2 + r^2}{k^2} \right).$$

Let, therefore, $$A = r^2 - k^2 \log \frac{k^2 + r^2}{k^2} \;\ldots\ldots\; (5)$$

$$\left.\begin{array}{ll} \text{the effective power} & = \tfrac{1}{2} A C \times v \\ \text{the full power} & = \tfrac{1}{2} A C \times u \end{array}\right\} \;\ldots\ldots\ldots\; (6)$$

Multiply by $\dfrac{60}{33000} = \dfrac{1}{550}$, and

the effective horse-powers $= \dfrac{1}{1100} A C \times v$, and

the full horse-powers $= \dfrac{1}{1100} A C \times u$.

Make $$c = \frac{W}{2g} \times \frac{\pi}{550} \;\ldots\ldots\ldots\ldots\; (7)$$

$$\left.\begin{array}{ll} \text{then the effective horse-powers} & = A c (u - v)^2 v, \\ \text{and the full horse-powers} & = A c (u - v)^2 u \end{array}\right\}. \;(8)$$

Again, by (6) the effective propelling pressure

$$= \tfrac{1}{2} A C = \frac{W}{2g} \pi A (u - v)^2.$$

But, if S = the effective surface of resistance of the vessel, the effective resistance $= \dfrac{W}{2g} \cdot S v^2$;

$$\therefore S v^2 = \pi A (u - v)^2,$$

$$\text{or} \quad \left(\frac{u}{v} - 1\right)^2 = \frac{S}{\pi A},$$

$$\left.\begin{array}{ll} \text{whence} & u = v\left(1 + \sqrt{\dfrac{S}{\pi A}}\right), \\ \text{the revolutions per second being} = \dfrac{u}{h} & \end{array}\right\} \quad \ldots\ldots\ldots (9)$$

Should the propelling surface of the screw be generated between two limiting valves r, r_1', it will be requisite to estimate the function A, formula (5), between those limits, *i. e.*,

$$A = \left(r^2 - k^2 \log \frac{k^2 + r^2}{k^2}\right) \sim \left(r_1^2 - k^2 \log \frac{k^2 + r_1^2}{k^2}\right). \; (10)$$

The proportion of the power of the engine usefully effective in the propulsion of the vessel

$$= \frac{\text{effective power}}{\text{full power}} = \frac{v}{u} = \frac{1}{1 + \sqrt{\dfrac{S}{\pi A}}},$$

$$\text{when} \quad S = \frac{2{\cdot}5\,(\text{tonnage})^{\frac{2}{3}}}{15} = \tfrac{1}{6}\,(\text{tonnage})^{\frac{2}{3}},$$

$$\text{or} \quad S = \frac{3\,(\text{tonnage})^{\frac{2}{3}}}{15} = \tfrac{1}{5}\,(\text{tonnage})^{\frac{2}{3}},$$

according to the form of the vessel.

That is, "till further experiments, and, consequently, better information is produced," says Mr. Woolhouse, "the effective resisting surface of the vessel may be estimated at about

$$\tfrac{1}{5} \text{ or } \tfrac{1}{6}\,(\text{tonnage})^{\frac{2}{3}}.$$

The larger the diameter and the less the pitch of the screw, the greater will be the proportion of the effective power on the vessel"

ON BOILERS.

A boiler for 20 horse-powers is usually 15 feet long and 6 feet wide; therefore 90 feet of surface, or $4\frac{1}{2}$ feet to 1 horse-power; a boiler of 14 horse-powers 60 feet of surface, or 4·3 feet to 1 horse-power; but engineers generally allow 5 feet of surface to 1 horse-power; and Mr. Hicks, of Bolton, proportions his boilers at the rate of $5\frac{1}{2}$ square feet of horizontal surface of water to each horse-power. Mr. Watt allows 25 cubic feet of space to each horse-power.

Length of Boilers for Locomotive Engines.

On the Northern and Eastern Counties Railway, the length of the boiler is 8 feet; while on the North Midland Counties Railway, on the Great Western Railway, and on several others, the length of the boiler is $8\frac{1}{2}$ feet. In Stephenson's locomotive engines, the length of the boiler is between 11 and 12 feet. On the Bordeaux and La Teste Railway the length of the boiler is 8 feet 9 inches; and in America the length varies from 10 to 14 feet.

The *Inside Diameter of a Locomotive Boiler* is usually found by multiplying the diameter of the cylinder in inches by $3\frac{1}{9}$, and the product is the inside diameter of the boiler. Thus, for example, let the diameter of the cylinder be 15 inches; then

$$15 \times 3\tfrac{1}{9} = \frac{15 \times 28}{9} = \frac{140}{3} = 46\tfrac{2}{3} \text{ inches},$$

the inside diameter of the boiler.

The *Inside Diameter of the Steam Dome* may obviously be varied considerably. It is usual, however, in practice to proportion its diameter to the diameter of the cylinder, by multiplying this diameter by 1·43 for the diameter of the steam dome in inches. Thus, for example, let the diameters of the cylinders be 15 inches; then

$$15 \times 1{\cdot}43 = 21{\cdot}45 \text{ inches} = \text{diameter of the dome.}$$

The *Height of the Steam Dome* may be considerably varied; but judging from practice, it appears that a uniform height of $2\frac{1}{2}$ feet would answer in all cases.

The Area of the Fire-grate, in practice, follows a more

certain rule than any other part of the engine seems to do; but it appears to be in all cases much too small, and causes a great loss of power by urging the blast it renders necessary, and a rapid deterioration of the furnace plates from excessive heat. There is no sufficient reason why the furnace should not be nearly as long as the boiler; it would then resemble the furnace of a marine engine boiler, and be quite as manageable. However, the practical rule to find the area of the fire-grate is to multiply the diameter of the cylinder in inches by ·77, and the product is the area of the fire-grate in feet. Thus, for example, let the diameter of the cylinders be 16 inches; then

$$16 \times \cdot 77 = 12 \cdot 32 \text{ square feet} = \text{area of the fire-grate.}$$

STEAM PIPES.

It is usual in practice to have the internal diameter of the steam pipe about one-fifth of the diameter of the cylinder of the engine. The area of the passages through the valves in some of Watt's engines are nearly one square inch to each horse-power. This is in some cases too large for steam passages, but rather too small for the exhausting-valve passages. Indeed, the larger the exhausting-valve passages are, the better.

The only proper method of proportioning the steam passages is by taking into consideration the velocity of the steam through them. Mr. Tredgold says, the force of steam in the boiler, multiplied by the velocity and the area of the passage, must be equal to the elastic force on the piston multiplied by its area and velocity. That is,

$$a \,.\, f \,.\, v = \mathrm{A\,V}\,p.$$

Where a is the area of the steam passages, f the force of the steam in the boiler in inches of mercury, v the velocity, p the force of the piston in the same denomination as f A the area of the piston, and V its velocity,

$$\therefore v = \frac{\mathrm{A\,V}\,p}{a \,.\, f},$$

$$\text{or} \quad a = \frac{\mathrm{A\,V}\,p}{f\,v}.$$

THE AIR-PUMP, CONDENSER, &c.

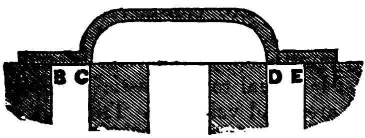
The proportion of the diameter of the air-pump, as given by Mr. Watt, is usually about two-thirds of the diameter of the cylinder, when the length of the stroke of the air-bucket is half the length of the stroke of the steam piston.

The area of the passages between the condenser and the air-pump should never be less than one-fourth of the area of the air-pump; the apertures through the air-bucket should have the same proportion, and, if convenient, the discharging flap or valve should be made larger.

The capacity of the condenser should at least be equal to that of the air-pump; and, where convenience will admit it, the larger it is made the better.

THE SLIDE-VALVE OF THE STEAM ENGINE.

When the breadth of the slide face of the valve is just equal to that of the steam port, the travel of the slide is evidently equal to twice the breadth of the steam port; but when the slide has lap, then the length of the stroke or travel of the slide is equal to double the length of the lap, together with double the breadth of the steam port. In the annexed figure, which represents the slide at half stroke, A B or E F



is the lap, and B C or D E the breadth of the steam port. When the slide begins to move from one extremity of its stroke, the point F arrives at D; and when it has passed over half its travel, which is its position as shown in the figure, it is evident that the point F has moved over the space F D, which is equal to the breadth of the steam port added to the lap; the slide next, moving through the other half of its travel, will bring the point A to C, a space which is also equal to the breadth of the steam port added to the lap; therefore the whole travel of the slide is equal to twice

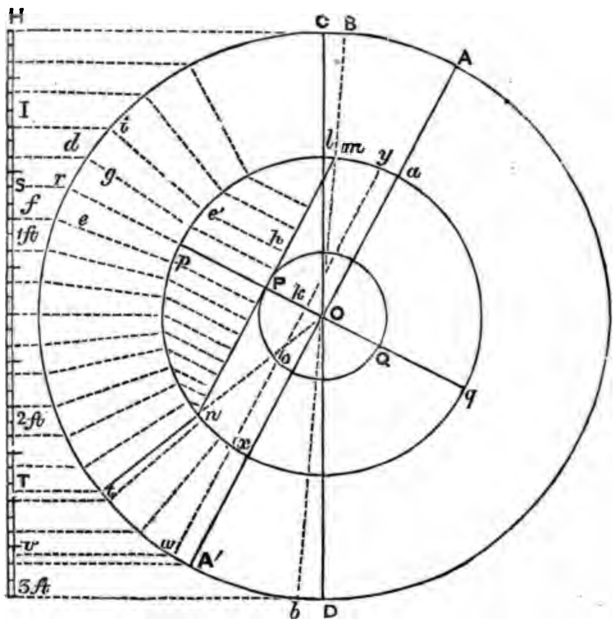
the sum of the breadth of the steam port and the length of the lap. In this figure the slide has no lead.

The following method of showing the positions of the piston, corresponding to the various openings of the steam and exhausting ports, has been presented to me for insertion in this work by the eminent engineer, Mr. Amos, of the well-known firm of Easton and Amos. It is given in as clear and simple a manner as the subject seems to admit of, since any intelligent working man may draw the figure to any convenient scale with rule and compasses, while, for locomotive engines, the figure might be drawn of the full size.

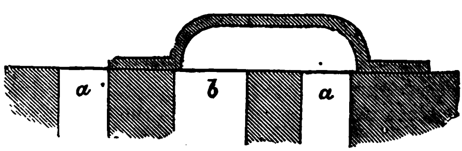
GEOMETRICAL CONSTRUCTION.

From a scale of equal parts take the length of the stroke of the piston, divided into feet and inches, and with radius equal to half the stroke describe the circle A C D, which will represent the crank orbit. Also form a scale of equal parts, the scale being as large as practicable; with the radius O P—equal to the lap of the slide—describe the circle *o* P Q; and with the radius O *p*—equal to the lap and breadth of one steam port—describe the circle *p q a*—the distance *p q* represents the travel of the slide; on this latter circle set off *l m* equal to the lead of the slide, and from *m* draw the line *m n*, touching the circle *o* P Q at P; and parallel to this line draw A A′, and A will represent the position of the crank-pin, when the slide is at half stroke. When the crank-pin arrives at B, the slide will begin to open; and when it arrives at C, the piston will be at the upper end of its stroke, and the steam port will be open a distance equal to *l m*, and taking the steam on the upper side of the piston to commence the down stroke. To ascertain the position of the crank and piston when the steam port is fully open, from *m* and *n* as centres, describe the arcs *d e* and *f g*, and their point of intersection *r*, in the circle A C D, will be the position of the crank-pin; and the line *r s*, parallel to C H, will be the position of the piston at the same time. When the crank-pin arrives at the point *t*, the slide will have returned to the line *m n*, and the steam will be cut off; and the line T *t*, drawn parallel to C H, will be the position of the piston, showing the portion of the half stroke performed. When the crank-pin arrives at *b*, the steam port begins to open for the admission of steam

to the under side of the piston, the crank having arrived at D.



To ascertain the position of the slide with respect to the piston at any portion of the stroke—as, for example, when the piston has performed 6 inches of its stroke—draw I *i* parallel to H C, and from *i* draw *i e'*, which if produced would pass through the centre O, and from the point *e'* draw *e' h* at right

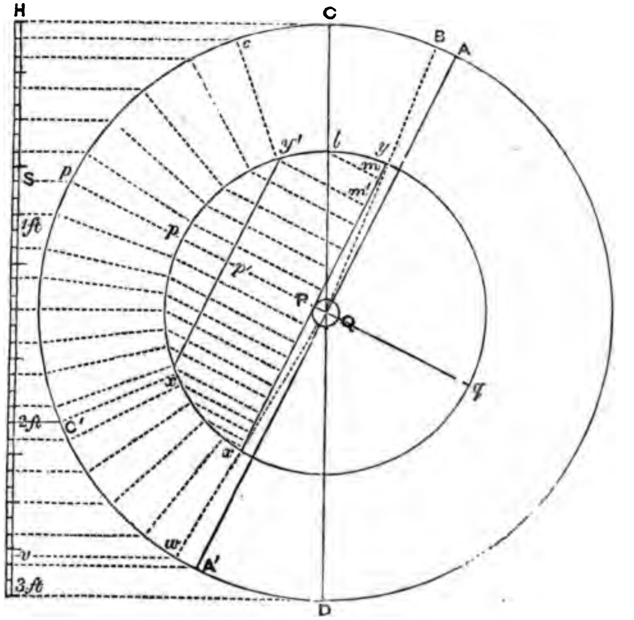


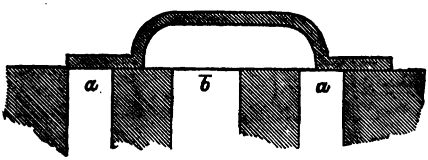
angles to *m n*; then the distance *e' h* will be equal to the breadth of the steam port which is open at that time. Again, *if the piston has travelled* 33½ inches, draw *v w* parallel to H C,

and from w draw $w\,x$ radiating to the centre O, and parallel to $m\,n$ draw $x\,y$, cutting p O in k; then the upper part will be closed, and the slide will have travelled the distance $p\,k$. See the preceding figure.

In the foregoing construction, the slide has been considered with respect to the admission of the steam only; we shall next consider it with respect to the exhaustion of the steam from the opposite side of the piston.

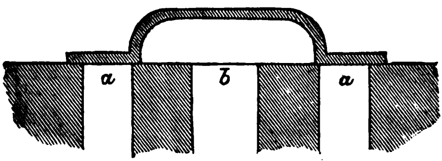
Draw the length of the stroke and the crank orbit as





before; also draw the line A A′, making the same angle with *C D, as before*; with the lap on the exhausting side as

radius, describe the circle $P\,o\,Q$; and parallel to the line $A\,A'$ draw the lines $x\,y$, $x'\,y'$, respectively cutting $p\,q$ in P and p'; then the distance $P\,p'$ will represent the breadth of the steam ports a to the cylinder (see the following figure of the slide), and p P will be equal to the breadth of the exhaust port b. When the crank is at A, the slide is at the middle of its stroke; when it arrives at B, the steam begins to exhaust from the under side of the piston; and when the crank arrives at C, the piston being at the same time at the upper end of its stroke, the slide will be opened the distance $l\,m$; and when the crank arrives at c, the same port will be open the distance $y'\,m'$, equal the width $P\,p'$ of the steam ports; when the crank arrives at r, the slide will be at the end of its travel, as shown in the preceding figure of the slide, p P being equal to the exhaust port b, and s will be the position of the piston in its down stroke. It will be seen that during the time the piston is travelling from c to c', the bottom port p' P has been fully open, to allow the steam to escape from the under side of the piston. When the crank arrives at w, the lower port is shut, and the slide is in the position shown in the following figure.



Formula and Examples for the Lap and Lead of the Slide.

Let l = lap, l' = lead, S = stroke of the piston, s = stroke of the slide, and d = distance before the steam is cut off; then

$$d = S\left\{1 - \left(\frac{2l + l'}{s}\right)^2\right\} \quad \ldots\ldots\ldots\ldots (1)$$

$$l = \tfrac{1}{2}\left\{s\sqrt{\frac{S - d}{S}} - l'\right\} \quad \ldots\ldots\ldots (2)$$

Example 1.—Given the lap = 1·85, the lead = ·2, the stroke of the piston = 40, and that of the slide = 6, all in inches; required the distance moved by the piston before the steam is cut off.

By formula (1),

$$40\left\{1-\left(\frac{2\times 1\cdot 85+\cdot 2}{6}\right)^2\right\}=40\ (1-\cdot 4225)=23\cdot 1 \text{ inches},$$

which is the distance the piston has moved before the steam was cut off.

Example 2.—The length of the stroke of the piston of an engine is 40, the steam is cut off at 23·1, the stroke of the slide is 6, and the lead $\frac{1}{5}$, all in inches; required the lap of the slide.

By formula (2),

$$\tfrac{1}{2}\left\{6\sqrt{\frac{40-23\cdot 1}{40}}-\cdot 2\right\}=\tfrac{1}{2}\,(3\cdot 9-\cdot 2)=1\cdot 85 \text{ inches},$$

the lap required.

THE LOCOMOTIVE ENGINE.

Descriptions having now been given of the general and peculiar principles of both the stationary and marine engines, it will next be proper to give a description of the peculiar properties of the locomotive engine, several of its most important general principles having necessarily already been given. This class of engines is adapted to either railways or common roads, but has been more successful on the former. The principle of action being the same in both kinds, a description of the railway variety will be sufficient, which will be given under the following eight heads:—

1st. The boiler, in which the steam as usual is generated, contains internally a fire-box, connected with several small tubes, varying in number from 50 to 300, a regulator, and a steam pipe. Externally, a chimney and two safety-valves are fixed to the boiler.

2nd. The parts which regulate the action of the steam are two slide-valves (covering the passages to and from the cylinders) attached to two sets of valve gear, worked by two eccentrics for the "forward," and two other eccentrics for

the "backward" motion of the engine; but only two of them work at one time, the other two being what is called "out of gear." Four rods, called eccentric rods, encircling the eccentric sheaves at one end, and jointed to the slide-valve gear at the other end, complete the connection of the slide-valves to the eccentrics fixed on the axle of the driving wheels.

3rd. The parts by which the engine-man controls the action of the locomotive are three sets of levers and rods, connected to the slide-valve, eccentric rods, regulator valves, and feed-pipe cocks; by which he can "put on" or "shut off" the steam to the cylinders, water to the boiler, put the slide-valves in a "backward" or "forward" position, at pleasure. These appendages are commonly called the *hand gear.*

4th. The parts essentially concerned in producing locomotion are two cylinders, in which work two steam-tight pistons with their piston rods. On the exterior ends of the piston rods are fixed T pieces, also called crossheads, which slide between or round guide bars, also called motion bars, fixed to the ends of, and parallel to, the cylinders. By this means the piston can only move in a right line in the same direction as the lengths of the cylinders. Two strong rods, called connecting rods, attach the crossheads to the arms of the driving wheels, or to a cranked axle, where there is one used. Whether the pistons are connected with a cranked axle or the arms of the driving wheels, these connections are always made at an angle of 45° to each other; therefore one piston is exerting its greatest force when the other is at the end of the cylinder, and exerting no force. The connection between the pistons and the driving wheels being thus fully completed, it is at once evident that any movement of the pistons must immediately act upon the driving wheels.

5th. The parts which supply water to the boiler are two forcing pumps, connected by two feeding pumps to the boiler, and to a reservoir of water in the tender to the locomotive. The pumps are worked either from the eccentrics on the axle of the driving wheels, or from the crosshead.

6th. The parts which excite the rapid combustion of fuel required in the locomotive engine are the chimney and the *blast-pipe, so contrived* as to cover the exhausting passages

from both cylinders, and terminating in the centre of the chimney, about the level of the top of the boiler. It is the escape through this pipe of each succeeding cylinder-full of steam, or that portion of it allowed to escape by the slide-valves, which causes the "beats" or "pulsations" so distinctly heard when the locomotive commences its motion. The more general properties and utility of the steam-blast will be detailed further on.

7th. The parts which support the locomotive engine are two, four, or six wheels, besides the two driving wheels, a suitable set of springs, and a strong frame, in which the machinery already described is securely fixed, and on which the boiler and cylinders are also securely bound.

8th. From the co-operation of the several parts already described locomotion is produced in the following manner The boiler is charged with water until it completely surrounds all the tubes and the internal fire-box, and fire being applied, in due time steam is produced from the water, and collected between the surface of the water and the top of the boiler until it has reached the pressure required. The regulator is then opened, and the slide-valves placed in their working position by the engine-man; the steam rushes from the boiler through the steam-pipe to the cylinders, where its force moves the pistons, which, being attached to the driving wheels (as has been already explained), cause them to revole, thus producing locomotion. The slide-valves and pumps, being wrought by some of the parts set in motion by the pistons, regulate the admission of steam to the cylinder, and of water to the boiler. When the steam has moved the piston to the end of the cylinder, a passage is opened for its escape to the atmosphere through the blast-pipe, and the intermitting velocity of this escaping steam creates a partial vacuum in the chimney, causing a rush or "blast" of air to fill this vacuum, which blast excites the rapid combustion of the fuel, and consequent rapid production of steam. This completes the duties of one admission of steam to the cylinder until its escape to the atmosphere, and when this escape has taken place, another admission of steam to the opposite side of the piston forces it back to the other end of the cylinder; and by means of the crank, the reciprocating motion of the piston is converted into a rotatory on. Thus the locomotion, begun by the first ad-

mission of steam to the cylinders, is continued by the second and succceeding admissions.

The repetition of these simple operations has amazed and gratified the world by safely conveying heavy passenger trains at upwards of sixty miles per hour, and merchandise trains of six hundred tons weight at twenty-five miles per hour! the mere idea of which, not many years since, would have been regarded as purely fabulous.

Such is the modern locomotive engine—an illustrative example of the genius of man; but, like other important inventions, it is the joint production of many minds, and many more are still directed to its further improvement. The records of the Patent Office show that, from January, 1840, to the end of September, 1849, no less than two hundred and twenty-six patents were enrolled, all of them more or less applicable to the steam engine and its appendages. Of these two hundred and twenty-six patents, forty-five were enrolled during the first nine months of 1849. It has been remarked that steam engines and railways were too matter-of-fact subjects for poets and painters, but from the above record it is evident that they deeply impress themselves upon the inventive intellect of the world; and if the prodigies performed by steam remain unsung, or unpourtrayed, they dare, if not realise, the very sublimity of both poetry and painting; for what scene more interesting to delineate than one of these stately machines moving safely along, at eagle-speed, the very *élite* of the land, including even royalty itself, through districts rich in historical associations of past ages, and at the same time teeming with the great works of Nature and Art! Surely, it cannot be that the subject is too lofty a one for poetical or pictorial illustration, for in greatness of idea lies the success of both.

A brief review of the progress of locomotives is all that can be here given. It is now (1862) about two thousand years since the powers of steam were shown and recorded by Hero of Alexandria; but it is little more than two hundred years (in 1650) since it was first usefully employed by the Marquis of Worcester. The first idea of using it for propelling carriages is generally ascribed to Dr. Robinson, in 1759, when it was suggested to him by Watt, who included a steam carriage in his patents of 1769 and 1784; but we *had no railways* then, and his patents were never carried

out, otherwise his scientific and comprehensive mind would speedily have accomplished what several of his less-talented successors were a long series of years in attaining. From 1802 to 1805, Trevithick applied steam carriages to both common roads and railways, with considerable success for first experiments; and his engines, with Booth's and Stephenson's modern improvements, now constitute the modern locomotive. About the year 1803, it appears that a Mr. Fredericks also made a steam locomotive for a silver mine in Hanover, which in 1811 was employed to convey their Majesties of Westphalia and suite over the mineral railway at a considerable speed. This was probably the first royal trip on a railway. From 1805 to 1814 invention was directed to ensure the adhesion of the wheels upon the rails, and many ingenious plans were tried, some of which succeeded well at slow speeds, but were not at all calculated for high velocities. In 1814, however, Mr. Blackett, of the Wylam Railway, reverting to Trevithick's plan, fully established the FACT, that on a level, or moderately inclined railway, the adhesion of a smooth iron wheel upon a smooth iron rail was sufficient to draw heavy loads; in his experiments he tried both six and eight-wheeled engines. In 1814 Mr. Stephenson introduced two cylinders, or two complete steam engines in one locomotive. From this time, up to 1829, the powerful opposition of the owners of other modes of conveyance greatly retarded the progress of locomotive engines; and so strong was the feeling that they were not economical, that both Mr. Walker and Mr. Rostrick reported against them in 1829. These reports, and one of a doubtful character by Telford, led to the offer of a prize of £500, in 1829, by the directors of the Liverpool and Manchester Railway, for the best locomotive engine, the weight of which was not to exceed six tons. This proceeding gave an important impulse to locomotives, and ended in establishing the superiority of railways over all other existing systems of travelling. Three competitors appeared, namely, Messrs. Stephenson, Erickson, and Hackworth; their locomotives were respectively named the *Rocket*, the *Novelty*, and the *Sanspareil*. They were all tried, and the prize was won by the *Rocket*, which, after the trials were over, reached a speed of thirty-five miles per hour. The other two engines reached the required speed per hour, as shown by their trials, but broke down before the required distance was completed.

The *Rocket* embraced the tubes and blast-pipe of the modern locomotive; the former (twenty-five in number) were suggested by Mr. Booth, of Liverpool, and the latter was first introduced by Mr. Hackworth, in his locomotives constructed previous to the *Sanspareil.*

The *Novelty* embraced a principle of generating a high heat by means of bellows, not since that time adopted in practice. She broke down after the trial commenced.

The *Sanspareil* embraced the blast-pipe (first introduced by her owner, Mr. Hackworth, as already stated), with the single return tube of the old locomotives. She was fully expected to accomplish the required distance greatly within the given time; but she unfortunately broke down, as already noticed, before the distance was fully completed.

It will hence be seen that Mr. G. Stephenson's *Rocket* won, chiefly through the adoption of the important suggestions of others, rather than through his own inventive genius. He can, therefore, *only* be considered a *fortunate* man, and Mr. Hackworth, a particularly *unfortunate* man, as his opponent, Mr. Stephenson, beat him by the help of the blast-pipe, which was Mr. Hackworth's own invention.

From 1830, up to the introduction of the seven-feet gauge on the Great Western Railway in 1838, no marked improvement took place in the locomotive. The rivalry, however, which sprung up between the gauges tended greatly to develop their capabilities.

A great number of patents have been enrolled for improving the locomotive engine, but only a few of these have been adopted in practice; among the most conspicuous of them is Mr. Crampton's arrangement of wheels, Mr. M'Connell's tank engine, Mr. Samuel's express engine, and Mr. Adam's steam carriage. The improvements in the mechanism of the slide-valve motion by Mr. Gray have been widely adopted. This great improvement is usually called Stephenson and Gray's, because the patent was so enrolled. Mr. Crampton has engines of his plan at work, both in England and on the Continent, which enable high driving wheels to be used on the narrow gauge without raising the centre of gravity. For illustrations of these, and other examples, see the new edition of "Tredgold on the Steam Engine."

The tank engine carries on the same frame water and fuel, *its tank for water being* placed on the top of the boiler.

This is the plan adopted on the Great Western Railway; but on the narrow gauge lines the tank is usually placed below the boiler and framing, which is a better arrangement where the machinery permits it to be done.

Mr. Samuel's express engine weighs only 25 cwt., and Mr. Adams' steam carriage is on this plan, with a very handsome carriage for passengers, all on one frame, and has been tried on some of the branch railways of both gauges.

Having thus briefly glanced at the progress of the locomotive engine, it only remains as briefly to notice some important discussions which have agitated the mechanical world concerning them.

From the earliest introduction of locomotives, four, six, or eight wheels appear to have been used, according to the designs of the makers; but about 1840-42 an animated discussion of the respective merits of the four and six-wheeled engines was carried on in the railway press. Both classes have their merits, and both had able advocates; but public opinion evidently tended in favour of the six-wheeled engine as the safer of the two under all contingencies; hence the greater portion of the present locomotives have six wheels.

The gauge controversy of 1845-48 led to the re-introduction of eight-wheeled engines on both gauges, weighing about 36 tons each, which realised speeds of about sixty and seventy miles per hour. The weight of these monster engines, it will be observed, was more than eight times the weight of the *Rocket* (4¼ tons), which won the prize in 1829; whilst the speed is only twice that of the *Rocket* (thirty-five miles per hour) at that time. It is worthy to remark that the existing engines in 1829 were from 10 to 16½ tons, and were considered as far too heavy; hence the directors of the Liverpool and Manchester Railway bound competitors not to exceed 6 tons weight. In 1849 the same feeling prevailed, and the injury done to the railways by these 36-tons engines was much complained of, and tank engines and steam carriages, till their weight was reduced, were equally objectionable; so that, generally, the weight of all locomotives is now reduced to what may be considered as proper and practicable in this important respect.

ON MOVING A TRAIN ON A LEVEL RAILWAY.

When the train of a locomotive engine commences its motion on a railway, the power of the engine exceeds the resistance, and therefore the speed increases until the resistance becomes equal to the power of the engine; the speed of the train will then be uniform, which is commonly called a steady speed, or the greatest or maximum speed; the work or motion destroyed by the resistance being now exactly equal to the power exerted by the engine. See formula (v), p. 32; see also "Statics and Dynamics," Weale's Series.

It has been found by experiment that the friction on a horizontal or inclined railway is invariably $= \frac{w}{n}$, or the nth part of the weight w of the train, $\frac{1}{n}$ being the co-efficient of friction; therefore the whole resistance to motion on the railway is also $= \frac{w}{n}$. Let P = power required to move the train, s = space in feet moved over in the time t in minutes, and H = number of horse-powers in P; then, since 33000 are the estimated number of units of work which one horse can perform in a minute, P = 33000 H = units of work or pounds moved one foot in one minute, and $\frac{s}{t} =$ feet moved in one minute by the weight $\frac{w}{n}$; whence $\frac{w}{n} \times \frac{s}{t}$ = units of work required in moving the carriages or train on a level railway.

$$\therefore \mathrm{P} = 33000\,\mathrm{H} = \frac{w}{n} \times \frac{s}{t},$$

whence $$\mathrm{H} = \frac{w\,s}{33000\,n\,t} \quad \ldots\ldots\ldots\ldots\ldots \text{(A)}$$

In railway calculations w is usually given in tons, and s in miles, which are to be reduced to pounds and feet by multiplying them respectively by 2240 and 5280; also n is most commonly = 280; if, therefore, we substitute 2240 W for w, 5280 S for s, and 280 for n, in formula (A), there will *result after reduction,*

$$H = \frac{128\ W\ .\ S}{100\ t} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

whence

$$W = \frac{100\ H\ .\ t}{128\ S} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

$$S = \frac{100\ H\ .\ t}{128\ W} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

and

$$t = \frac{128\ W\ .\ S}{100\ H} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

Example 1.—Required the horse-powers (H) of a locomotive engine, which moves with a steady speed of 50 miles per hour on a level railway, the weight of the train being 45 tons, and the friction $\frac{1}{280}$ of the weight of the train, the resistance of the air not being considered.

By formula (1),

$$H = \frac{128\ W\ .\ S}{100\ t} = \frac{128 \times 45 \times 50}{100 \times 60} = 48 \text{ horse-powers.}$$

Example 2.—An engine of 40 horse-powers moves with a steady speed of 35 miles per hour on a level railway; required the weight of the train, the friction being as usual.

By formula (2),

$$W = \frac{100\ H\ .\ t}{128\ S} = \frac{100 \times 40 \times 60}{128 \times 35} = 53\tfrac{4}{7} \text{ tons.}$$

Example 3.—In what time will an engine of 50 horse-powers, moving a train of 60 tons, complete a distance of 80 miles?

By formula (4),

$$t = \frac{128\ W\ .\ S}{100\ H} = \frac{128 \times 60 \times 80}{100 \times 50} = 122\tfrac{22}{25} \text{ min.} = 2 \text{ h. } 2\tfrac{22}{25} \text{ min.}$$

Example 4.—How many miles per hour will a train of 40 tons be drawn by an engine of 35 horse-powers?

By formula (3),

$$S = \frac{100 \times H\ .\ t}{128\ W} = \frac{100 \times 35 \times 60}{128 \times 40} = 41\tfrac{1}{64} \text{ miles per hour.}$$

Example 5.—At what rate per hour will a train of 100 tons be drawn on a level railway by an engine of 50 horse-powers?

Example 6.—How many pounds per ton does the friction amount to, in Example 1, the engine being of 48 horse-powers?

By transposing formula (A),

$$\frac{1}{n} = \frac{33000 \, . \, t \, . \, H}{w \, . \, s} = \frac{33000 \times 60 \times 48}{2240 \times 45 \times 5280 \times 50} = \frac{1}{280} \text{ of}$$

the weight of the train, or 8 lbs. per ton, the values of w and s being reduced to pounds and feet respectively.

ON MOVING A TRAIN AGAINST THE JOINT RESISTANCES OF FRICTION AND GRAVITY ON AN INCLINED PLANE.

Let P = power, and w = weight in pounds of a train, and h = rise of the inclined railway in every 100 feet of its length; then by Art. (66), Form. (2), Baker's "Statics and Dynamics," Weale's Series, $P = \frac{100 + h\,n}{100\,n} \cdot w$; and let H, s, and t respectively represent the horse-powers, space in feet, and time required in moving the weight $\frac{100 + h\,n}{100\,n} \cdot w$, as in the last articles; then P = 33000 H = units of work in pounds, $\frac{s}{t}$ = feet moved in one minute by the weight $\frac{100 + h\,n}{100\,n} \cdot w$; whence $\frac{100 + h\,n}{100\,n} \cdot w \times \frac{s}{t}$ = units of work required in moving the weight w, which must be equal to the units of work in the power;

$$\therefore P = 33000 = \frac{100 + h\,n}{100\,n} \cdot w \times \frac{s}{t};$$

whence

$$H = \frac{(100 + h\,n)\,s\,w}{33000 \times 100\,n\,t} \quad \text{............} \quad (B)$$

Now, let W = weight moved in tons, and S the space or distance moved in miles, as usually given in railway calculations; then $w = 2240$ W, and $s = 5280$ S; these values

being substituted in formula (B,) and n being taken $= 280$, as in the last article, there will result after reduction,

$$H = \frac{256\,(5 + 14\,h)\,W\,.\,S}{1000\,t} \quad \text{.........} \quad (1)$$

whence $$W = \frac{1000\,t\,.\,H}{256\,(5 + 14\,h)\,S} \quad \text{...............} \quad (2)$$

$$S = \frac{1000\,t\,.\,H}{256\,(5 + 14\,h)\,W} \quad \text{...............} \quad (3)$$

$$t = \frac{256\,(5 + 14\,h)\,W\,.\,S}{1000\,H} \quad \text{.........} \quad (4)$$

and $$h = \frac{1000\,t\,.\,H - 1280\,W\,.\,S}{3584\,W\,.\,S} \quad \text{......} \quad (5)$$

In all these formulæ h must be taken negatively when the train moves on a descending gradient of a railway, in which case gravity assists the moving power. It also appears that when h is negative and equal to $\frac{5}{14}$ of a foot, then no power is required to move the train, for the value of H vanishes, since in this case $5 + 14\,h = 0$.

Example 1.—A train of 40 tons ascends a railway gradient, rising 2 feet in 100, with a uniform speed of 15 miles per hour; required the horse-powers of the locomotive engine, the friction being as usual.

By formula (1),

$$H = \frac{256\,(5 + 14\,h)\,W\,.\,S}{1000\,t} = \frac{256\,(5 + 28) \times 40 \times 15}{1000 \times 60}$$
$$= 84\tfrac{12}{25} \text{ horse-powers.}$$

Example 2.—Required the horse-powers, as in the last example, when the weight of the train is 60 tons, the rise $\frac{1}{2}$ foot in 100, and the rate of motion 30 miles per hour.

Answer.—92 horse-powers.

Example 3.—An engine of 75 horse-powers ascends a gradient, rising $\frac{3}{4}$ of a foot in 100, with a uniform speed of 20 miles per hour; required the weight of the train.

By formula (2),

$$W = \frac{1000\,t\,.\,H}{256\,(5 + 14\,h)\,S} = \frac{1000 \times 60 \times 75}{256\,(5 + \frac{21}{2}) \times 20} = 56{\cdot}7 \text{ tons.}$$

Example 4.—A train of 120 tons descends a gradient, falling $\frac{1}{4}$ of a foot in 100, with a uniform speed of 50 miles per hour; what are the horse-powers exerted by the engine?

Here the h must be negative, because the train descends the gradient; hence, by formula (1),

$$H = \frac{256\ (5 - 14\,h)\ S\ .\ W}{1000\,t} = \frac{256\ (5 - \frac{7}{2}) \times 50 \times 120}{1000 \times 60}$$

$$= 38\tfrac{2}{5} \text{ horse-powers.}$$

Example 5.—A train of 50 tons ascends a gradient, having a rise of $\frac{1}{6}$ of a foot in 100; required the speed of the engine when its horse-powers are 40.

SAFETY VALVES.

In locomotive engines there are two safety valves placed on the boiler for the escape of steam when it exceeds the pressure limited by the load on these valves. One of them is placed beyond the control of the engine-man, and is commonly called the *lock-up valve.* The other valve is regulated by a lever and spring, at a somewhat lower pressure than that on the lock-up valve. The apertures for safety valves require no nice calculations. It is only necessary to have the aperture sufficiently large to let the steam off from the boiler as fast as it is generated, when the engine is not at work.

The safety valve of a locomotive engine is sometimes loaded by putting a heavy weight upon it in the case of the lock-up valve, and sometimes the other valve is secured by means of a lever with a weight to move along it to suit the required pressure, as in the stationary engine, which has already been explained, and the formula for graduating the lever given.

ON THE VARIABLE RESISTANCES TO LOCOMOTIVE ENGINES.

This engine (besides the friction of the rims of the wheels on the rails, which has been already the subject of investigation) has to overcome other variable resistances, viz., that

of the atmosphere, which varies according to the direction of the wind, which, sometimes, when very high, and in the same direction as the motion of the engine, assists its speed. The direct reverse takes place when a high wind is a-head of the direction of the motion of the engine. All other directions of the winds assist or retard its motion more or less, except direct cross-winds, which may be said to be nearly neutral in their effects. There is also a somewhat variable resistance from the friction of the axles, from the moving parts of the engine, and from the passage of the steam through the blast-pipe.

The resistance of the atmosphere is difficult to determine; it is considered to vary as the square of the velocity. The result of the experiments of Pambour on these subjects shall here be given, and also examples to illustrate the theory.

The power of the locomotive engine cannot be exactly estimated by the pressure of steam in the boiler, and the diameter and length of the stroke of the piston, since the elastic force of the steam is diminished, in passing from the boiler to the cylinders, by the smallness of the apertures of the steam-pipes through which it has to pass. This diminution is also frequently produced by the evaporating power of the boiler not being capable of keeping up a supply of steam to the cylinders of the same elasticity as that in the boiler; hence the pressure upon the piston is less than that upon the safety valve of the boiler. This diminution of the steam in the cylinders, compared with that in the boiler, will frequently be in the ratio of the increase of the velocity of the engine. Thus, supposing an engine can evaporate a certain quantity of water per hour, of the elasticity shown by the valve on the boiler; if this production of steam is sufficient to supply as many cylinders-full of steam, of the same density as that in the boiler, as shall be equal to the number of strokes per minute of the piston required to produce the given velocity, the elasticity of the steam in the cylinders will be very nearly equal to that in the boiler, and, consequently, the pressure on the piston will also be very nearly equal to that in the boiler. But if the velocity of the engine is such that the number of cylinders-full of steam required is greater than the evaporation of the boiler can supply at the elasticity shown by the safety valve, then the elasticity in the cylinders is proportionally diminished.

Example.—A railway train moves at the rate of 30 miles per hour upon a level rail, the resistance from friction is 8 lbs. per ton, the resistance of the atmosphere 30 lbs. on the train when the rate is 10 miles per hour, the diameter of the driving wheels 6 feet, the area of the piston 100 square inches, the length of the stroke 18 inches, the resistance due to the blast-pipe is $1\frac{3}{4}$ lbs. per square inch of the piston when the rate is 10 miles per hour; required the pressure of the steam, the evaporation of the boiler, the number of bushels of coals for a journey of 150 miles, supposing that one bushel of coals can evaporate 11 cubic feet of water, the weight of the train being 90 tons.

$8 \times 90 = 720 =$ resistance from the friction,

$\left(\frac{30}{10}\right)^2 \times 30 = 270 =$ resistance of the atmosphere against the train.

$720 + 270 = 990$ lbs. $=$ whole resistance to the motion of the train,

$6 \times 3{\cdot}1416 = 18{\cdot}8496$ feet moved over by the driving wheels in one revolution,

$18{\cdot}8496 \times 990 = 18661{\cdot}104 =$ work of resistance in one revolution.

Since the engine has two cylinders, and each piston makes two strokes in one revolution of the driving wheels, the work done in one stroke must be multiplied by 4; hence,

$1 \times 100 \times \frac{3}{2} \times 4 = 600 =$ the work of 1 lb. pressure per square inch on the pistons in one revolution of the driving wheels.

The effective pressure of one square inch of the piston, multiplied by the work of 1 lb. per square inch pressure in one revolution, must be equal to the resistances in one revolution; hence,

$\frac{18661{\cdot}104}{600} = 31{\cdot}102$ lbs. $=$ pressure on one square inch.

From the experiments of Pambour, it has been found *that the resistance* of the blast-pipe, or steam-jet, increases

directly as the velocity, and the co-efficient he gives is ·175; and hence, at any given velocity it is ·175 multiplied by that velocity, that is, for a velocity of 30 miles per hour.

$\cdot 175 \times 30 = 5{\cdot}25$ lbs. = resistance of the blast-pipe,

$\therefore 31{\cdot}102 + \frac{1}{7} \times 31{\cdot}102 + 5{\cdot}25 + 15 + 1 = 56{\cdot}795$ lbs. = pressure per square inch.

$\frac{30 \times 5280}{6 \times 3{\cdot}1416 \times 60} = 140 =$ number of revolutions of the driving wheels per minute.

$4 \times 140 = 560 =$ number of strokes of the piston per minute.

$\frac{100}{144} \times \frac{3}{2} \times 560 = 583$ cubic feet = volume of steam discharged per minute; but by the table one cubic foot of water produces 498 cubic feet of steam at 56 lbs. pressure (the decimal ·795 being neglected); hence the number of cubic feet evaporated per minute is $\frac{583}{498} = 1{\cdot}17$; and since one bushel of coals evaporates 11 cubic feet of water, the number of bushels of coals used in one minute is $= \frac{1{\cdot}17}{11}$, and for 5 hours, which it requires to go over 150 miles, there results $\frac{1{\cdot}17}{11} \times 60 \times 5 = 32$ bushels of coals, nearly, to convey 90 tons 150 miles. For further information on these important subjects, see the new edition of "Tredgold on the Steam Engine."

Note.—In concluding the subject of the motion of locomotive engines on railways, it will be here proper to remark that the methods of laying out railway-curves on the ground, of cutting, embanking, &c., were first prepared by the author of this work at the commencement of the railway era (1824); and have since that time been fully adopted in practice, not only in this kingdom and its numerous colonies, but also in India, the United States, and other foreign countries; more than thirty thousand copies of his works, embodying these methods in a practical form, have been sold. He also gave a greatly improved method of calculating the contents of railway cuttings, and remodelled the several methods of dividing lands and commons of variable

value among the several claimants, in proportion to the value of their claims, thus giving these important parts of applied mathematics a more scientific and comprehensive form than they previously possessed. See Baker's "Land and Engineering Surveying," Weale's Series, and Baker's "Engineering and Earthwork," and Nugent's edition of Railway Construction, Weale's Series, No. 62.

ON THE STEAM-BLAST, OR STEAM-JET, IN LOCOMOTIVES, ETC.

The steam-jet in locomotive engines leads from the exhaust passages of the cylinders into the chimney, and causes the draught through the fire-tubes necessary for the enormous consumption of fuel which is required in the locomotive, since each jet of steam emitted creates a partial vacuum in the chimney, which is immediately filled by a current of air rushing through the fire-grate.

"This mode of draught, which is at the same time economical and of extreme simplicity, is preferable to all others, inasmuch as, by using it, the draught may be varied more easily than by making use of machines, the effect of which cannot increase the draught, except by an augmentation of the velocity, which is frequently found inconvenient and very limited. It would be important to bring the steam to the exhausting ports by a pipe of larger diameter, in order that in its passage the steam should experience but little friction, and also that the exhausting ports should have the proper dimensions, so that the steam at its issue might nearly acquire the velocity which corresponds to the pressure in the boiler. It has been asserted that, in the steam-jet, all the effect produced by the *vis viva* of the steam was not obtained unless the air was at a temperature sufficiently high for no condensation of the steam to take place; but it appears that this circumstance bears no very sensible influence. However, it would be useful to try some experiments on that subject."

"It has also been asserted, from experiments made on locomotives, that by the use of an intermittent jet a greater useful effect was produced than by a continuous one. Supposing this to be true, it should be seen if, in the experiments that have been made, the tension of the steam was not greater with the intermittent jet than by the continuous one." See M. Peclet's "Treatise on Heat," Art. 577

ON CHIMNEYS.

From numerous experiments, the following laws are deduced in reference to the production of the motion of air or gases, in pipes or close galleries :—

1st. All other things being the same, the pressure required to generate velocity only is proportional to the square of such velocity.

2nd. All other things being the same, the pressure required to overcome the resistance offered by pipes, walls, or sides of galleries is also proportional to the square of the velocity with which the air moves.

Whence it follows that, all other things being equal, whatever may be the pressure and velocity causing ventilation, a certain fixed proportion or percentage is expended in generating the velocity, and the remaining constant proportion is expended in overcoming the frictional resistance by the wall of the chimney.

From the experiments of Regnault, it appears that air expands 0·3665 of its volume at 32° of Fahrenheit, by being heated from that temperature to 212° on the same scale; hence the friction

$$\frac{0{\cdot}3665}{212^\circ - 32} = \frac{1}{491}$$

has been adopted as the correct measure of the expansion of air for each degree of Fahrenheit between the limits just stated.

Let $v =$ the volume of any given weight of air at the temperature of 32°, $v' =$ the volume it assumes on being heated to any higher temperature t, and $v'' =$ the volume at the temperature t'; then

$$v' : v'' :: v\left(1 + \frac{t - 32}{491}\right) : v\left(1 + \frac{t' - 32}{491}\right),$$

whence $$v''\left(1 + \frac{t - 32}{491}\right) = v'\left(1 + \frac{t' - 32}{491}\right),$$

from which

$$v'' = \left(\frac{t' + 491 - 32}{t + 491 - 32}\right) v' = \left(\frac{t' + 459}{t + 459}\right) v' \ldots (1)$$

an equation from which the volume of air at any given temperature can readily be found from its volume at any other temperature.

It is known that if h = height of a chimney in which air is allowed to expand freely by heat, under a constant pressure (barometrical); by heating the air in such a chimney from a temperature t to a higher temperature t', its contents will be expanded in the ratio of

$$h \text{ to } h\left(\frac{t' + 459}{t + 459}\right);$$

and hence the difference,

$$h\left(\frac{t' + 459}{t + 459}\right) - h, \text{ or } h\left(\frac{t' + 459}{t + 459} - 1\right) = \frac{h\,(t' - t)}{t + 459} \quad \ldots \; (2)$$

would represent the length of a column of air at t', having the same area as the chimney, which would be expelled from it by such an increase of temperature. The height of such expelled column, reduced to air of the temperature t, would be

$$\frac{h\,(t' - t)}{t + 459} \div \frac{t' + 459}{t + 459} = \frac{h\,(t' - t)}{t' + 459} \quad \ldots\ldots \; (3)$$

This may be called the *motive height;* whence

$$v = 8\sqrt{\textit{motive height}} = 8\sqrt{h\left(\frac{t' - t}{t' + 459}\right)} \quad \ldots \; (4)$$

which is the same theorem as that given by Tredgold, excepting the constant, which is a little less than that given by Regnault.

MISCELLANEOUS EXAMPLES.

1. Given the area of the piston of a high-pressure stationary engine 400 square inches, the length of the stroke 6 feet, the evaporation of the boiler $1\frac{1}{2}$ cubic feet of water per minute, and the pressure of steam in the cylinder 60 lbs. per square inch; required the useful load and the horse-powers.

2. In a condensing engine the area of the cylinder is 1000 square inches, the length of the stroke including clearance 5 feet, the steam is cut off at 1 foot of the stroke, the clearance is $\frac{1}{4}$ of a foot, the pressure of steam is 30 lbs., the elasticity of vapour in the condenser is 4 lbs., the effective evaporation of the boiler is $\frac{1}{5}$ of a cubic foot per minute, and the resistance as usual; required the useful load and horse-powers. See solution to question 3, page 33.

3. Let the area of the piston be 1800 square inches, the length of the stroke including clearance $10\frac{1}{2}$ feet, the clearance being $\frac{1}{2}$ of a foot, the steam is cut off at 2 feet of the stroke, the pressure of steam in the cylinder is 48 lbs. per square inch, the elasticity of vapour in the condenser is 4 lbs., the total resistance from friction is $1\frac{3}{4}$ lbs. per square inch of the piston, the effective evaporation of the boiler is $\frac{9}{10}$ of a cubic foot per minute; required the useful work per minute, the effective horse-powers, the number of strokes per minute, and the duty of the engine, allowing one bushel to evaporate 10 cubic feet of water.

4. The area of the piston is 1500 square inches, length of the stroke including clearance of 6 inches is 9 feet, the steam is cut off at 3 feet of the stroke, the pressure of steam in the cylinder is 45 lbs. per square inch, the elasticity of the vapour in the condenser is 3 lbs., and the resistance is $1\frac{1}{2}$ lbs., and the number of strokes per minute is 20; required the quantity of water evaporated per minute, the horse-powers, and the useful load on the piston.

5. The area of the piston is 1800 square inches, $\frac{19}{20}$ of a cubic foot of water is evaporated per minute, the number of strokes per minute is 18, the length of the stroke including clearance of 6 inches is $10\frac{1}{2}$ feet, the steam is cut off at 2 feet of the stroke, and the total resistance of vapour in the condenser and of friction are together $5\frac{1}{2}$ lbs.; required the horse-powers.

6. The horse-powers of an engine are 200, the length of stroke with clearance of 6 inches is 10 feet, the steam is cut off at 20 inches of the stroke, the pressure of the steam is 45 lbs., the elasticity of vapour in the condenser is 4 lbs., the resistance from friction 2 lbs. per square inch, and the number of strokes 20; required the area of the piston, and the useful load on each square inch thereof.

7. The area of the piston is 4000 square inches, the length of the stroke including clearance of $\frac{1}{2}$ a foot is $10\frac{1}{2}$ feet, the pressure of the steam 40 lbs. per square inch, the pressure of vapour in the condenser and the resistance from friction are together 5 lbs. per square inch; required at what part of the stroke the steam must be cut off so as to yield all its work, the horse-powers of the engine, the useful load on the piston, and the quantity of water evaporated.

8. Let the area of the piston = 3600 square inches, the length of the stroke = 10 feet, the number of strokes per minute = 18, the elasticity of the steam = 36 lbs., that of the vapour in the condenser including the resistance of friction 5 lbs., and the steam is cut off at $\frac{1}{8}$ of the stroke; required the horse-powers of the engine, the load, and the point at which the velocity of the piston is a maximum.

9. The length of the stroke is 12 feet, the pressure of steam in the cylinder is 50 lbs.; at what point of the stroke must the steam be cut off so as to yield all its work, when the resistance of the vapour in the condenser together with the friction of the engine is 5 lbs. per square inch on the piston?

10. The length of the stroke is 10 feet, the steam is cut off at $\frac{1}{8}$ of the stroke, the area of the piston is 3600 square inches, the number of strokes per minute is 18, the clearance is $\frac{1}{2}$ a foot, the pressure of vapour in the condenser is 3 lbs., and the water evaporated per minute is $\frac{3}{4}$ of a cubic foot; required the useful load, supposing the friction of the unloaded piston to be 5 lbs. per square inch, and the additional friction to be $\frac{1}{8}$ of the useful load.

11. The area of the piston is 2000 square inches, the length of the stroke including the clearance of 6 inches is 12 feet, the effective evaporation is $\frac{19}{20}$ of a cubic foot per minute, the pressure of the steam in the cylinder is 50 lbs., and the elasticity of the vapour in the condenser together with the resistance of friction is $6\frac{1}{2}$ lbs.; required the point at which *the steam* must be cut off, so that all its work may be

expended, the number of strokes per minute, the useful work, the effective horse-powers, and the duty of the engine, allowing that one bushel of coals can evaporate 12 cubic feet of water.

12. The diameter of the cylinder is 40 inches, the length of the stroke 10 feet, the clearance $\frac{1}{20}$ of the stroke, the pressure in the boiler 21·5 lbs. per square inch, the effective evaporation $\frac{9}{10}$ of a cubic foot of water per minute, and the consumption of coals in the same time $8\frac{1}{2}$ lbs.; required the useful horse-powers, when the piston moves with the velocity of 250 feet per minute; also find the useful effects of one pound of coals and one cubic foot of water.

Note.—The solution to all these questions may be obtained by means of the formulæ, commencing at page 26.

APPENDIX.

ON THE STRENGTH, FRICTION, ETC., OF SEVERAL IMPORTANT PARTS OF THE STEAM ENGINE.

ON GUDGEONS.

In gudgeons, one-fifth of the diameter is usually allowed for wear, and Mr. Tredgold, on this principle, gives the following

RULE.—Multiply the stress in pounds by the length of the gudgeon in inches, and the cube root of the product, divided by 9, is the diameter of the gudgeon in inches.

Example.—If the stress on a gudgeon be 12 tons, and its length 8 inches; required its diameter.

$$2240 \times 12 \times 8 = 215040$$

$$\sqrt[3]{215040} = 60 \text{ nearly,}$$

$$\text{and } \frac{60}{9} = 6\tfrac{2}{3} \text{ inches, the diameter of the gudgeon.}$$

CRANKS.

The force or weight acting on a crank being given, to find its breadth and depth.

RULE.—Multiply the weight in pounds, acting at the end of a crank, by the cube of its length in feet, and this product being divided by 2662 times the deflection, will give the product of the cube of the depth and the breadth in inches.

Example.—If the force acting upon a crank be 6000 lbs., and its length 3 feet; required its breadth and depth, so that the deflection may not exceed $\frac{1}{10}$ of an inch.

By the rule, $\frac{6000 \times 3^3}{2662 \times \cdot 1} = 609$ nearly = cube of the depth multiplied by the breadth.

If the breadth be assumed = 3 inches, then

$$\sqrt[3]{\frac{609}{3}} = 6 \text{ inches nearly} = \text{the depth.}$$

If the depth at the end where the force acts be half the depth at the axis, divide by 1628 instead of 2662; then

$\frac{6000 \times 3^3}{1628 \times \cdot 1} = 995 =$ cube of the depth multiplied by the breadth; and if the breadth be taken = 3 inches, as before, then

$$\sqrt[3]{\frac{995}{3}} = 6{\cdot}93 \text{ inches, the greater depth;}$$

$$\text{and } \frac{6{\cdot}93}{2} = 3{\cdot}465 \text{ inches, the lesser depth.}$$

FRICTION OF AN AXLE.

Let P = pressure on the bearings,
r = radius of the axle,
n = number of revolutions per second,
f = ratio of the friction to the pressure corresponding to the bodies in contact,
and π = 3·1416; then

$$2\pi \, . \, P \, . \, f \, . \, r \, . \, n = \text{work of friction per second.}$$

Example.—Given the radius of the axle = 6 inches, the weight of the shaft and other parts pressing on it = 1500 lbs., the shaft making 10 revolutions per minute; required the work expended on friction.

Let the co-efficient of friction f = ·07, and since 6 inches = ·5 of a foot, there results from the preceding formula

$$\frac{2 \times 3{\cdot}1416 \times 1500 \times \cdot 07 \times \cdot 5 \times 10}{60} = 54{\cdot}978 \text{ lbs.,}$$

the quantity of work expended on friction per second.

FRICTION ON A PIVOT.

The symbols P, r, n, f, representing the same things as in the last article, there will result the following formula for the friction on a pivot,

$$4 \cdot 2 \times P \cdot f \cdot r \cdot n$$

= the work expended on friction per second.

Example.—Given the radius of the base of the pivot = 3 inches, the weight pressing upon it 2000 lbs.; the pivot makes 10 revolutions per minute; required the work expended on friction.

Let the co-efficient of friction $f = \cdot 07$, and since 3 inches $= \cdot 25$ of a foot, there results from the formula

$$\frac{4 \cdot 2 \times 2000 \times \cdot 07 \times \cdot 25 \times 10}{60} = 24\tfrac{1}{2} \text{ lbs. nearly.}$$

FRICTION OF CYLINDERS.

The proportion which the friction of a large cylinder bears to the friction of any number of cylinders, the sum of the areas of which is equal to the area of the large cylinder, may be shown in the following manner:—

Let d = diameter of one of the small cylinders,
and n = the number of them; then

$$\frac{\pi}{4} n d^2 = \text{area of the large cylinder;}$$

$$\therefore d \sqrt{n} = \text{diameter of the large cylinder.}$$

But the friction is proportional to the circumference of the cylinder; therefore the friction of the small cylinders is

$$\pi d n,$$

and the friction of the large cylinder is

$$\pi d \sqrt{n}.$$

Hence, *the friction of the large cylinder* : *the friction of all the small cylinders* :: $\pi d \sqrt{n} : \pi d n$,

$$:: \quad 1 \quad : \sqrt{n};$$

which proportion, in the case of four small cylinders, becomes as 1 : 2, that is, the friction of four cylinders is double the friction of one cylinder, the area of which is equal to the areas of all the four.

ON WINDING MACHINES,

MOVED BY THE STEAM ENGINE, FOR DRAWING COALS OUT OF A SHAFT.

In winding machines of this kind the steam engine is intended to produce a given number of strokes in drawing up a corf; hence, the diameter of the roll must be ascertained at the first lift. In this case it is supposed that the machine has flat ropes, such as are now generally used, and which coil upon each other.

To find the diameter of the roll at the first lift, it will be necessary to know the thickness of the rope, the depth of the shaft or pit, and the number of strokes which the steam engine is intended to make in drawing up a corf or corves.

Then the number of strokes of the engine, and the thickness of the rope being both known, the thickness of the rope upon the roll can be determined, let the diameter of the roll be what it may. Thus, suppose the thickness of the rope be 1 inch, and the number of strokes 12; the radius of the roll is increased 12 inches, or the diameter is increased 24 inches, whatever that diameter may be.

Put $d =$ the depth of the shaft or pit,
and $t =$ the thickness of the rope, both in inches;
also $n =$ the number of strokes of the engine,
$\pi = 3{\cdot}1416$,
and $x =$ the diameter of roll.

Then $$n = \frac{d - \pi n^2 t}{\pi n}.$$

Example.—If an engine makes 20 strokes in drawing a corf up a shaft, the depth of which is 100 fathoms, and the thickness of the rope 1 inch; required the diameter of the roll at the first lift.

By the formula,

$$x = \frac{d - \pi n^2 t}{\pi n};$$

that is, $x = \frac{100 \times 72 - 3{\cdot}1416 \times 20^2 \times 1}{3{\cdot}1416 \times 20} =$

$$\frac{7200 - 1256{\cdot}64}{62{\cdot}832} = 94{\cdot}59 \text{ inches} =$$

7 feet 10·6 inches nearly, the required diameter of the roll.

It is proper here to remark that if an engine be drawing coals out of a pit where round ropes have been used, and if it be required to take the round ones off, and supply their place with flat ones, the formula just given will determine the diameter of the roll at the first lift, so that the engine may go the same number of strokes as when the round ropes were on.

Example.—If an engine goes 10 strokes in drawing a corf up a shaft, the depth of which is 60 fathoms, with round ropes, where these round ropes do not coil upon each other, what must be the diameter of the flat rope roll, so that the engine may go the same number of strokes as before, the thickness of the rope being ½ an inch?

By the formula,

$$x = \frac{d - \pi n^2 t}{\pi n},$$

that is, $x = \frac{60 \times 72 - 3{\cdot}1416 \times 10^2 \times \frac{1}{2}}{3{\cdot}1416 \times 10} =$

$$\frac{4320 - 157{\cdot}08}{31{\cdot}416} = 132{\cdot}5 \text{ inches} =$$

11 feet 0½ inch = the diameter of the rope roll.

To find the Position of the Meetings in a Coal Shaft.

When an engine draws coals out of a shaft with flat ropes, the corves will not pass each other at mid-shaft, that is, half way between the top and bottom of the shaft; for the *corf which* descends from the top of the shaft will pass

through a greater space in the same time than the corf which ascends from the bottom, owing to the circumference of the roll being always greater until the engine has performed half its number of strokes; therefore the meetings will always be below mid-shaft. At the meetings the number of coils of the ropes on the roll are equal, and after this the roll on which the ascending corf hangs continues to increase until the corf arrives at the top of the shaft.

*To find where the Ascending and Descending Corves will meet.**

Let the depth of the shaft and the thickness of the rope be represented as in the last formula, and let r = radius of the roll, and let the distance of the meeting from the bottom of the shaft = y; then we may use either of the following formulæ:—

$$y = \tfrac{1}{4}\,\pi\,n^2\,t + \pi\,r\,n.$$

$$y = d - \tfrac{3}{4}\,\pi\,n^2\,t - \pi\,n\,r. \qquad y = \tfrac{1}{2}\,d - \tfrac{1}{4}\,\pi\,n^2\,t.$$

Example.—At what distance from the bottom of a shaft will the corves meet, the following being given?

$d = 600$ feet, $t = \frac{1}{12}$ foot, $n = 20$, $r = 3{\cdot}9418$ feet, as in last problem.

Calculation by the first Formula.

$$y = 3{\cdot}1416 \times 100 \times \tfrac{1}{12} + 31{\cdot}416 \times 3{\cdot}9418 \times 2 = 26{\cdot}18 + 247{\cdot}63976 = 273{\cdot}82.$$

Hence, the distance of the place of meeting from the bottom of the pit is 274 feet, within about two inches.

Calculation by the second Formula.

$$y = 600 - \tfrac{1}{16} \times 400 - 31{\cdot}416 \times 3{\cdot}9418 \times 2 = 600 - 78{\cdot}54 - 247{\cdot}64 = 273{\cdot}82.$$

Calculation by the third Formula.

$$y = 300 - 3{\cdot}1416 \times 100 \times \tfrac{1}{12} = 300 - 314{\cdot}16 \times \tfrac{1}{12} = 300 - 26{\cdot}18 = 273{\cdot}82.$$

Hence, as before, the place of meeting is at the distance 273·82 feet, or 274 feet *nearly*, from the bottom of the pit.

* The formulæ and investigations following have been kindly supplied by Professor J. R. Young.

Demonstrations of the Formulæ in the last Problem.

In investigating formulæ for the foregoing problem, it must be remembered that when the corves meet, the number of coils on the roll for the ascending corve is the same as the number for the descending corve, the engine having then performed half its number of strokes. Hence, putting y for the distance of the place of meeting from the bottom of the pit, and regarding y, for the moment, as the entire depth, the number of strokes for this depth being $\frac{1}{2} n$, we have

$$2 r = \frac{y - \frac{1}{4} \pi n^2 t}{\frac{1}{2} \pi n}$$

$\therefore r \pi n = y - \frac{1}{4} \pi n^2 t$, and therefore $y = \frac{1}{4} \pi n^2 t + \pi r n$;

which is the first of the formulæ employed.

Again: the diameter of the coil at the first lift is $2 r$; and when the corf is drawn up the distance y from the bottom of the pit, the engine having performed $\frac{1}{2} n$ strokes, $\frac{1}{2} n t$ expresses the thickness added to the coil, or the amount by which the radius r is increased; in other words, when the corf is at the distance y from the bottom, the *diameter* of the coil is $2 r + n t$, which, of course, is the same as the diameter of the coil for the other corf, when this is at the distance $d - y$ from the top. Hence,

$$\frac{d - y - \frac{1}{4} \pi n^2 t}{\frac{1}{2} \pi n} = 2 r + n t$$

$$\therefore d - y - \frac{1}{4} \pi n^2 t = \pi n r + \frac{1}{2} \pi n^2 t$$

$$\therefore y = d - \frac{1}{2} \pi n^2 t - \frac{1}{4} \pi n^2 t - \pi n r = d - \frac{3}{4} \pi n^2 t - \pi n r;$$

which is the second formula employed; and the third is half the sum of these two. A *fourth* formula is $y = \frac{1}{2} d + \frac{1}{2} \pi r n$.

NOTE TO THE TABLE OF HYPERBOLIC LOGARITHMS.—The common log. of any number multiplied by 2·30258505 gives the hyperbolic log.; and the hyperbolic log. multiplied by ·43429448 gives the common log.—EDITOR.

TABLE OF HYPERBOLIC LOGARITHMS.

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
1·01	·0099503	1·34	·2926696	1·67	·5128236
1·02	·0198026	1·35	·3001045	1·68	·5187937
1·03	·0295588	1·36	·3074846	1·69	·5247285
1·04	·0392207	1·37	·3148107	1·70	·5306282
1·05	·0487902	1·38	·3220834	1·71	·5364933
1·06	·0582689	1·39	·3293037	1·72	·5423242
1·07	·0676586	1·40	·3364722	1·73	·5481214
1·08	·0769610	1·41	·3435897	1·74	·5538851
1·09	·0861777	1·42	·3506568	1·75	·5596157
1·10	·0953102	1·43	·3576744	1·76	·5653138
1·11	·1043600	1·44	·3646431	1·77	·5709795
1·12	·1133287	1·45	·3715635	1·78	·5766133
1·13	·1222176	1·46	·3784364	1·79	·5822156
1·14	·1310283	1·47	·3852624	1·80	·5877866
1·15	·1397619	1·48	·3920420	1·81	·5933268
1·16	·1484200	1·49	·3987761	1·82	·5988365
1·17	·1570037	1·50	·4054651	1·83	·6043159
1·18	·1655144	1·51	·4121097	1·84	·6097655
1·19	·1739533	1·52	·4187103	1·85	·6151856
1·20	·1823215	1·53	·4252677	1·86	·6205764
1·21	·1906203	1·54	·4317824	1·87	·6259384
1·22	·1988508	1·55	·4382549	1·88	·6312717
1·23	·2070141	1·56	·4446858	1·89	·6365768
1·24	·2151113	1·57	·4510756	1·90	·6418538
1·25	·2231435	1·58	·4574248	1·91	·6471032
1·26	·2311117	1·59	·4637340	1·92	·6523251
1·27	·2390169	1·60	·4700036	1·93	·6575200
1·28	·2468600	1·61	·4762341	1·94	·6626879
1·29	·2546422	1·62	·4824261	1·95	·6678293
1·30	·2623642	1·63	·4885800	1·96	·6729444
1·31	·2700271	1·64	·4946962	1·97	·6780335
1·32	·2776317	1·65	·5007752	1·98	·6830968
1·33	·2851789	1·66	·5068174	1·99	·6881346
1·34	·2926696	1·67	·5128236	2·00	·6931472

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
2·01	·6981347	2·34	·8501509	2·67	·9820784
2·02	·7030974	2·35	·8544153	2·68	·9858167
2·03	·7080357	2·36	·8586616	2·69	·9895411
2·04	·7129497	2·37	·8628899	2·70	·9932517
2·05	·7178397	2·38	·8671004	2·71	·9969486
2·06	·7227059	2·39	·8712933	2·72	1·0006318
2·07	·7275485	2·40	·8754687	2·73	1·0043015
2·08	·7323678	2·41	·8796267	2·74	1·0079579
2·09	·7371640	2·42	·8837675	2·75	1·0116008
2·10	·7419373	2·43	·8878912	2·76	1·0152306
2·11	·7466879	2·44	·8919980	2·77	1·0188473
2·12	·7514160	2·45	·8960880	2·78	1·0224509
2·13	·7561219	2·46	·9001613	2·79	1·0260415
2·14	·7608058	2·47	·9042181	2·80	1·0296194
2·15	·7654678	2·48	·9082585	2·81	1·0331846
2·16	·7701082	2·49	·9122826	2·82	1·0367368
2·17	·7747271	2·50	·9162907	2·83	1·0402766
2·18	·7793248	2·51	·9202827	2·84	1·0438040
2·19	·7839015	2·52	·9242589	2·85	1·0473189
2·20	·7884573	2·53	·9282193	2·86	1·0508216
2·21	·7929925	2·54	·9321640	2·87	1·0543120
2·22	·7975071	2·55	·9360933	2·88	1·0577902
2·23	·8020015	2·56	·9400072	2·89	1·0612564
2·24	·8064758	2·57	·9439058	2·90	1·0647107
2·25	·8109302	2·58	·9477893	2·91	1·0681530
2·26	·8153648	2·59	·9516578	2·92	1·0715836
2·27	·8197798	2·60	·9555114	2·93	1·0750024
2·28	·8241754	2·61	·9593502	2·94	1·0784095
2·29	·8285518	2·62	·9631743	2·95	1·0818051
2·30	·8329091	2·63	·9669838	2·96	1·0851892
2·31	·8372475	2·64	·9707789	2·97	1·0885619
2·32	·8415671	2·65	·9745596	2·98	1·0919233
2·33	·8458682	2·66	·9783261	2·99	1·0952733
2·34	·8501509	2·67	·9820784	3·00	1·0986123

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
3·01	1·1019400	3·34	1·2059707	3·67	1·3001916
3·02	1·1052568	3·35	1·2089603	3·68	1·3029127
3·03	1·1085626	3·36	1·2119409	3·69	1·3056264
3·04	1·1118575	3·37	1·2149127	3·70	1·3083328
3·05	1·1151415	3·38	1·2178757	3·71	1·3110318
3·06	1·1184149	3·39	1·2208299	3·72	1·3137236
3·07	1·1216775	3·40	1·2237754	3·73	1·3164082
3·08	1·1249295	3·41	1·2267122	3·74	1·3190856
3·09	1·1281710	3·42	1·2296405	3·75	1·3217558
3·10	1·1314021	3·43	1·2325605	3·76	1·3244189
3·11	1·1346227	3·44	1·2354714	3·77	1·3270749
3·12	1·1378330	3·45	1·2383742	3·78	1·3297240
3·13	1·1410330	3·46	1·2412685	3·79	1·3323660
3·14	1·1442227	3·47	1·2441545	3·80	1·3350010
3·15	1·1474024	3·48	1·2470322	3·81	1·3376291
3·16	1·1505720	3·49	1·2499017	3·82	1·3402504
3·17	1·1537315	3·50	1·2527629	3·83	1·3428648
3·18	1·1568811	3·51	1·2556160	3·84	1·3454723
3·19	1·1600209	3·52	1·2584609	3·85	1·3480731
3·20	1·1631508	3·53	1·2612978	3·86	1·3506671
3·21	1·1662709	3·54	1·2641266	3·87	1·3532544
3·22	1·1693813	3·55	1·2669475	3·88	1·3558351
3·23	1·1724821	3·56	1·2697605	3·89	1·3584091
3·24	1·1755733	3·57	1·2725655	3·90	1·3609765
3·25	1·1786549	3·58	1·2753627	3·91	1·3635373
3·26	1·1817271	3·59	1·2781521	3·92	1·3660916
3·27	1·1847899	3·60	1·2809338	3·93	1·3686394
3·28	1·1878434	3·61	1·2837077	3·94	1·3711807
3·29	1·1908875	3·62	1·2864740	3·95	1·3737156
3·30	1·1939224	3·63	1·2892326	3·96	1·3762440
3·31	1·1969482	3·64	1·2919836	3·97	1·3787661
3·32	1·1999647	3·65	1·2947271	3·98	1·3812818
3·33	1·2029722	3·66	1·2974631	3·99	1·3837912
3·34	1·2059707	3·67	1·3001916	4·00	1·3862943

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
4·01	1·3887912	4·34	1·4678743	4·67	1·5411590
4·02	1·3912818	4·35	1·4701758	4·68	1·5432981
4·03	1·3937663	4·36	1·4724720	4·69	1·5454325
4·04	1·3962446	4·37	1·4747630	4·70	1·5475625
4·05	1·3987168	4·38	1·4770487	4·71	1·5496879
4·06	1·4011829	4·39	1·4793292	4·72	1·5518087
4·07	1·4036429	4·40	1·4816045	4·73	1·5539252
4·08	1·4060969	4·41	1·4838746	4·74	1·5560371
4·09	1·4085449	4·42	1·4861396	4·75	1·5581446
4·10	1·4109869	4·43	1·4883995	4·76	1·5602476
4·11	1·4134230	4·44	1·4906543	4·77	1·5623462
4·12	1·4158531	4·45	1·4929040	4·78	1·5644405
4·13	1·4182774	4·46	1·4951487	4·79	1·5665304
4·14	1·4206957	4·47	1·4973883	4·80	1·5686159
4·15	1·4231083	4·48	1·4996230	4·81	1·5706971
4·16	1·4255150	4·49	1·5018527	4·82	1·5727739
4·17	1·4279160	4·50	1·5040774	4·83	1·5748464
4·18	1·4303112	4·51	1·5062971	4·84	1·5769147
4·19	1·4327007	4·52	1·5085119	4·85	1·5789787
4·20	1·4350845	4·53	1·5107219	4·86	1·5810384
4·21	1·4374626	4·54	1·5129269	4·87	1·5830939
4·22	1·4398351	4·55	1·5151272	4·88	1·5851452
4·23	1·4422020	4·56	1·5173226	4·89	1·5871923
4·24	1·4445632	4·57	1·5195132	4·90	1·5892352
4·25	1·4469189	4·58	1·5216990	4·91	1·5912739
4·26	1·4492691	4·59	1·5238800	4·92	1·5933085
4·27	1·4516138	4·60	1·5260563	4·93	1·5953389
4·28	1·4539530	4·61	1·5282278	4·94	1·5973653
4·29	1·4562867	4·62	1·5303947	4·95	1·5993875
4·30	1·4586149	4·63	1·5325568	4·96	1·6014057
4·31	1·4609379	4·64	1·5347143	4·97	1·6034198
4·32	1·4632553	4·65	1·5368672	4·98	1·6054298
4·33	1·4655675	4·66	1·5390154	4·99	1·6074358
4·34	1·4678743	4·67	1·5411590	5·00	1·6094379

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
5·01	1·6114359	5·34	1·6752256	5·67	1·7351891
5·02	1·6134300	5·35	1·6770956	5·68	1·7369512
5·03	1·6154200	5·36	1·6789639	5·69	1·7387102
5·04	1·6174060	5·37	1·6808278	5·70	1·7404661
5·05	1·6193882	5·38	1·6826882	5·71	1·7422189
5·06	1·6213664	5·39	1·6845453	5·72	1·7439687
5·07	1·6233408	5·40	1·6863989	5·73	1·7457155
5·08	1·6253112	5·41	1·6882491	5·74	1·7474951
5·09	1·6272778	5·42	1·6900958	5·75	1·7491998
5·10	1·6292405	5·43	1·6919395	5·76	1·7509374
5·11	1·6311994	5·44	1·6937790	5·77	1·7526720
5·12	1·6331544	5·45	1·6956155	5·78	1·7544036
5·13	1·6351056	5·46	1·6974487	5·79	1·7561323
5·14	1·6370530	5·47	1·6992786	5·80	1·7578579
5·15	1·6389967	5·48	1·7011051	5·81	1·7595805
5·16	1·6409365	5·49	1·7029282	5·82	1·7613002
5·17	1·6428726	5·50	1·7047481	5·83	1·7630170
5·18	1·6448050	5·51	1·7065646	5·84	1·7647308
5·19	1·6467336	5·52	1·7083778	5·85	1·7664416
5·20	1·6486586	5·53	1·7101878	5·86	1·7681496
5·21	1·6505798	5·54	1·7119944	5·87	1·7698546
5·22	1·6524974	5·55	1·7137979	5·88	1·7715567
5·23	1·6544112	5·56	1·7155981	5·89	1·7732559
5·24	1·6563214	5·57	1·7173950	5·90	1·7749523
5·25	1·6582280	5·58	1·7191887	5·91	1·7766458
5·26	1·6601310	5·59	1·7209792	5·92	1·7783364
5·27	1·6620303	5·60	1·7227666	5·93	1·7800242
5·28	1·6639260	5·61	1·7245507	5·94	1·7817091
5·29	1·6658182	5·62	1·7263316	5·95	1·7833912
5·30	1·6677068	5·63	1·7281094	5·96	1·7850704
5·31	1·6695918	5·64	1·7298840	5·97	1·7867469
5·32	1·6714733	5·65	1·7316555	5·98	1·7884205
5·33	1·6733512	5·66	1·7334238	5·99	1·7900914
5·34	1·6752256	5·67	1·7351891	6·00	1·7917594

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
6·01	1·7934247	6·34	1·8468787	6·67	1·8976198
6·02	1·7950872	6·35	1·8484547	6·68	1·8991179
6·03	1·7967470	6·36	1·8500283	6·69	1·9006138
6·04	1·7984040	6·37	1·8515994	6·70	1·9021075
6·05	1·8000582	6·38	1·8531680	6·71	1·9035989
6·06	1·8017098	6·39	1·8547342	6·72	1·9050881
6·07	1·8033586	6·40	1·8562979	6·73	1·9065751
6·08	1·8050047	6·41	1·8578592	6·74	1·9080600
6·09	1·8066481	6·42	1·8594181	6·75	1·9095425
6·10	1·8082887	6·43	1·8609745	6·76	1·9110228
6·11	1·8099267	6·44	1·8625285	6·77	1·9125011
6·12	1·8115621	6·45	1·8640801	6·78	1·9139771
6·13	1·8131947	6·46	1·8656293	6·79	1·9154509
6·14	1·8148247	6·47	1·8671761	6·80	1·9169226
6·15	1·8164520	6·48	1·8687205	6·81	1·9183921
6·16	1·8180767	6·49	1·8702625	6·82	1·9198594
6·17	1·8196988	6·50	1·8718021	6·83	1·9213247
6·18	1·8213182	6·51	1·8733394	6·84	1·9227877
6·19	1·8229351	6·52	1·8748743	6·85	1·9242486
6·20	1·8245493	6·53	1·8764069	6·86	1·9257074
6·21	1·8261608	6·54	1·8779371	6·87	1·9271641
6·22	1·8277699	6·55	1·8794650	6·88	1·9286186
6·23	1·8293763	6·56	1·8809906	6·89	1·9300710
6·24	1·8309801	6·57	1·8825138	6·90	1·9315214
6·25	1·8325814	6·58	1·8840347	6·91	1·9329696
6·26	1·8341801	6·59	1·8855533	6·92	1·9344157
6·27	1·8357763	6·60	1·8870696	6·93	1·9358598
6·28	1·8373699	6·61	1·8885837	6·94	1·9373017
6·29	1·8389610	6·62	1 8900954	6·95	1·9387416
6·30	1·8405496	6·63	1·8916048	6·96	1·9401794
6·31	1·8421356	6·64	1·8931119	6·97	1·9416152
6·32	1·8437191	6·65	1·8946168	6·98	1·9430489
6·33	1·8453002	6·66	1·8961194	6·99	1·9444805
6·34	1·8468787	6·67	1·8976198	7·00	1·9459101

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
7·01	1·9473376	7·34	1·9933387	7·67	2·0373166
7·02	1·9487632	7·35	1·9947002	7·68	2·0386195
7·03	1·9501866	7·36	1·9960599	7·69	2·0399207
7·04	1·9516080	7·37	1·9974177	7·70	2·0412203
7·05	1·9530275	7·38	1·9987736	7·71	2·0425181
7·06	1·9544449	7·39	2·0001278	7·72	2·0438143
7·07	1·9558604	7·40	2·0014800	7·73	2·0451088
7·08	1·9572739	7·41	2·0028305	7·74	2·0464016
7·09	1·9586853	7·42	2·0041790	7·75	2·0476928
7·10	1·9600947	7·43	2·0055258	7·76	2·0489823
7·11	1·9615022	7·44	2·0068708	7·77	2·0502701
7·12	1·9629077	7·45	2·0082140	7·78	2·0515563
7·13	1·9643112	7·46	2·0095553	7·79	2·0528408
7·14	1·9657127	7·47	2·0108949	7·80	2·0541237
7·15	1·9671123	7·48	2·0122327	7·81	2·0554049
7·16	1·9685099	7·49	2·0135687	7·82	2·0566845
7·17	1·9699056	7·50	2·0149030	7·83	2·0579624
7·18	1·9712993	7·51	2·0162354	7·84	2·0592388
7·19	1·9726911	7·52	2·0175661	7·85	2·0605135
7·20	1·9740810	7·53	2·0188950	7·86	2·0617866
7·21	1·9754689	7·54	2·0202221	7·87	2·0630580
7·22	1·9768549	7·55	2·0215475	7·88	2·0643278
7·23	1·9782390	7·56	2·0228711	7·89	2·0655961
7·24	1·9796212	7·57	2·0241929	7·90	2·0668627
7·25	1·9810014	7·58	2·0255131	7·91	2·0681277
7·26	1·9823798	7·59	2·0268315	7·92	2·0693911
7·27	1·9837562	7·60	2·0281482	7·93	2·0706530
7·28	1·9851308	7·61	2·0294631	7·94	2·0719132
7·29	1·9865035	7·62	2·0307763	7·95	2·0731719
7·30	1·9878743	7·63	2·0320878	7·96	2·0744290
7·31	1·9892432	7·64	2·0333976	7·97	2·0756845
7·32	1·9906103	7·65	2·0347056	7·98	2·0769384
7·33	1·9919754	7·66	2·0360119	7·99	2·0781907
7·34	1·9933387	7·67	2·0373166	8·00	2·0794415

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
8·01	2·0806907	8·34	2·1210632	8·67	2·1598687
8·02	2·0819384	8·35	2·1222615	8·68	2·1610215
8·03	2·0831845	8·36	2·1234584	8·69	2·1621729
8·04	2·0844290	8·37	2·1246539	8·70	2·1633230
8·05	2·0856720	8·38	2·1258479	8·71	2·1644718
8·06	2·0869135	8·39	2·1270405	8·72	2·1656192
8·07	2·0881534	8·40	2·1282312	8·73	2·1667653
8·08	2·0893918	8·41	2·1294214	8·74	2·1679101
8·09	2·0906287	8·42	2·1306098	8·75	2·1690536
8·10	2·0918640	8·43	2·1317967	8·76	2·1701959
8·11	2·0930984	8·44	2·1329822	8·77	2·1713367
8·12	2·0943306	8·45	2·1341664	8·78	2·1724763
8·13	2·0955613	8·46	2·1353491	8·79	2·1736146
8·14	2·0967905	8·47	2·1365304	8·80	2·1747517
8·15	2·0980182	8·48	2·1377104	8·81	2·1758874
8·16	2·0992444	8·49	2·1388889	8·82	2·1770218
8·17	2·1004691	8·50	2·1400661	8·83	2·1781550
8·18	2·1016923	8·51	2·1412419	8·84	2·1792868
8·19	2·1029140	8·52	2·1424163	8·85	2·1804172
8·20	2·1041341	8·53	2·1435893	8·86	2·1815467
8·21	2·1053529	8·54	2·1447609	8·87	2·1826747
8·22	2·1065702	8·55	2·1459312	8·88	2·1838015
8·23	2·1077861	8·56	2·1471001	8·89	2·1849207
8·24	2·1089998	8·57	2·1482681	8·90	2·1860512
8·25	2·1102128	8·58	2·1494339	8·91	2·1871742
8·26	2·1114243	8·59	2·1505987	8·92	2·1882959
8·27	2·1126343	8·60	2·1517622	8·93	2·1894163
8·28	2·1138428	8·61	2·1529243	8·94	2·1905355
8·29	2·1150499	8·62	2·1540851	8·95	2·1916535
8·30	2·1162555	8·63	2·1552445	8·96	2·1927702
8·31	2·1174596	8·64	2·1564026	8·97	2·1938856
8·32	2·1186622	8·65	2·1575593	8·98	2·1949998
8·33	2·1198634	8·66	2·1587147	8·99	2·1961128
8·34	2·1210632	8·67	2·1598687	9·00	2·1972245

Nos.	Logarithm.	Nos.	Logarithm.	Nos.	Logarithm.
9·01	2·1983350	9·34	2·2343062	9·67	2·2690282
9·02	2·1994443	9·35	2·2353763	9·68	2·2700618
9·03	2·2005523	9·36	2·2364452	9·69	2·2710944
9·04	2·2016591	9·37	2·2375130	9·70	2·2721258
9·05	2·2027647	9·38	2·2385797	9·71	2·2731562
9·06	2·2038691	9·39	2·2396452	9·72	2·2741856
9·07	2·2049722	9·40	2·2407096	9·73	2·2752142
9·08	2·2060741	9·41	2·2417729	9·74	2·2762411
9·09	2·2071748	9·42	2·2428350	9·75	2·2772673
9·10	2·2082744	9·43	2·2438960	9·76	2·2782924
9·11	2·2093727	9·44	2·2449559	9·77	2·2793165
9·12	2·2104697	9·45	2·2460147	9·78	2·2803395
9·13	2·2115656	9·46	2·2470723	9·79	2·2813614
9·14	2·2126603	9·47	2·2481288	9·80	2·2823823
9·15	2·2137538	9·48	2·2491843	9·81	2·2834022
9·16	2·2148461	9·49	2·2502386	9·82	2·2844211
9·17	2·2159372	9·50	2·2512917	9·83	2·2854389
9·18	2·2170272	9·51	2·2523438	9·84	2·2864556
9·19	2·2181160	9·52	2·2533948	9·85	2·2874714
9·20	2·2192034	9·53	2·2544446	9·86	2·2884861
9·21	2·2202898	9·54	2·2554934	9·87	2·2894998
9·22	2·2213750	9·55	2·2565411	9·88	2·2905124
9·23	2·2224590	9·56	2·2575877	9·89	2·2915241
9·24	2·2235418	9·57	2·2586332	9·90	2·2925347
9·25	2·2246235	9·58	2·2596776	9·91	2·2935443
9·26	2·2257040	9·59	2·2607209	9·92	2·2945529
9·27	2·2267833	9·60	2·2617631	9·93	2·2955604
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TABLE OF FRICTION OF GUDGEONS OR AXLE ENDS IN MOTION ON THEIR BEARINGS.

(From the experiments of M. Morin.)

Surfaces in Contact.	State of the Surface.	Co-efficients of Friction. In the usual way.	Co-efficients of Friction. Continuously.
Cast-iron axles in cast-iron bearings	greased with oil of olives, hog's lard, tallow, or soft gom . . .	0·07 to 0·08	0·054
	with the same and water	0·075	0·28
	coated with asphaltum .	0·054	0·19
	greasy	0·14	
	greasy and wetted . .	0·13	
Cast-iron axles in wrought - iron bearings . . .	greased with oil of olives, hog's lard, tallow, or soft gom	0·07 to 0·08	0·056
	greasy	0·16	
	greasy and damped . .	0·16	
	scarcely greasy . . .	0·19	
Cast-iron axles in lignum - vitæ bearings . . .	without unguent . . .	0·18	
	with oil or hog's lard .	—	0·09
	greasy with ditto . .	0·10	
	greasy with a mixture of hog's lard and molybdena	0·14	
Wrought-iron axles in cast-iron bearings	greased with oil of olives, hog's lard, tallow, or soft gom	0·07 to 0·08	0·054
Iron axles in brass bearings	greased with oil of olives, hog's lard, or tallow .	0·07 to 0·08	0·054
	coated with hard gom .	0·09	
	greasy and wetted . .	0·19	
	scarcely greasy . . .	0·25	
Iron axles in lignum-vitæ bearings	greased with oil, or hog's lard	0·11	
	greasy	0·19	
Brass axles in brass bearings . . .	greased with oil . . .	0·10	
	with hog's lard . . .	0·09	
Brass axles in cast-iron bearings .	greased with oil, or tallow	—	0·045 to 0·052
Lignum-vitæ axles in ditto . . .	greased with hog's lard .	0·12	
	greasy	0·15	
Lignum-vitæ axles in lignum-vitæ bearings . . .	greased with hog's lard .	—	0·07

TABLE OF FRICTION OF PLANE SURFACES IN MOTION ONE UPON THE OTHER.

(From the experiments of M. Morin.)

Surfaces in Contact.	Disposition of the Fibres.	State of the Surfaces.	Co-efficient of Friction.
Oak upon oak	parallel	without unguent	0·48
	ditto	rubbed with dry soap	0·16
	perpendicular	without unguent	0·34
	ditto	steeped in water	0·25
	wood endways on wood lengthways	without unguent	0·19
Elm upon oak	parallel	ditto	0·43
	perpendicular	ditto	0·45
Ash, fir, beech, wild pear-tree, and service-tree upon oak	ditto	ditto	0·36 to 0·40
Iron upon oak	ditto	ditto	0·62
		with water	0·26
		rubbed with dry soap	0·21
		without unguent	0·49
Cast-iron upon oak	ditto	with water	0·26
		rubbed with dry soap	0·19
Copper upon oak	ditto	without unguent	0·62
Iron upon elm	ditto	ditto	0·25
Cast-iron upon elm	ditto	ditto	0·20
Black dressed leather upon oak	ditto	ditto	0·27
Tanned leather upon oak	lengthways	ditto	0·32
		with water	0·29
Tanned leather upon cast-iron and brass	ditto	without unguent	0·56
		steeped in water	0·36
		greased and steeped in water	0·23
		with oil	0·15

Surfaces in Contact.	Disposition of the Fibres.	State of the Surfaces.	Co-efficient of Friction.
Hemp, in threads or in cord, upon oak	parallel	without unguent	0·52
	perpendicular	with water	0 33
Oak and elm upon cast-iron	parallel	without unguent	0·38
Wild pear-tree ditto	ditto	ditto	0·44
Iron upon iron	ditto	ditto	0·44
Iron upon cast-iron and brass	ditto	ditto	0·18
Cast-iron ditto	ditto	ditto	0·15
Brass upon brass	ditto	ditto	0·20
Brass upon cast-iron	ditto	ditto	0·22
Brass upon iron	ditto	ditto	0·16
Oak, elm, yoke elm, wild pear, cast-iron, wrought-iron, steel, moving one upon another, or on themselves	ditto	greased in the usual way, with tallow, hog's lard, oil, or soft gom	0·07 to 0·08
		slightly greasy to the touch	0·15
Calcareous oölite stone upon calcareous oölite	ditto	without unguent	0·64
Calcareous stone, called muschelkalk, upon calcareous oölite	ditto	ditto	0·67
Common brick upon calcareous oölite	ditto	ditto	0·65
Oak upon calcareous oölite	wood endways	ditto	0·38
Wrought-iron ditto	parallel	ditto	0·69
Calcareous stone, called muschelkalk, upon muschelkalk	ditto	ditto	0·38
Calcareous oölite stone upon muschelkalk	ditto	ditto	0·65
Common brick ditto	ditto	ditto	0·60
Oak upon muschelkalk	wood endways	ditto	0·38
Iron upon muschelkalk	parallel	ditto	0·24
		saturated with water	0·30

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